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TRANSMITTAL FORM (to be used for all correspondence after initial filing)		
	Application Number	10/016,472
	Filing Date	December 10, 2001
	First Named Inventor	Anthony J. Grzesiak et al.
	Art Unit	3683
	Examiner Name	Melody M. Burch
Total Number of Pages in This Submission	Attorney Docket Number	DKT 00065A (BWI-00056)

ENCLOSURES (Check all that apply)

- [x] Fee Transmittal Form
  - [x] Fee Attached
- [ ] Amendment/Reply
  - [ ] After Final
  - [ ] Affidavits/declaration(s)
- [ ] Extension of Time Request
- [ ] Express Abandonment Request
- [ ] Information Disclosure Statement
- [ ] Certified Copy of Priority Document(s)
- [ ] Response to Missing Parts/ Incomplete Application
  - [ ] Response to Missing Parts under 37 CFR 1.52 or 1.53
- [ ] Drawing(s)
- [ ] Licensing-related Papers
- [ ] Petition
- [ ] Petition to Convert to a Provisional Application
- [ ] Power of Attorney, Revocation Change of Correspondence Address
- [ ] Terminal Disclaimer
- [ ] Request for Refund
- [ ] CD, Number of CD(s) ____________
- [ ] After Allowance communication to Group
- [ ] Appeal Communication to Board of Appeals and Interferences
- [x] Appeal Communication to Group **(Appeal Notice, Brief, Reply Brief)**
- [ ] Proprietary Information
- [ ] Status Letter
- [x] Other Enclosure(s) (please Identify below):

Check for $330.00; and Return Receipt Postcard

Remarks

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SIGNATURE OF APPLICANT, ATTORNEY, OR AGENT

Firm or Individual name	Warn, Burgess & Hoffmann, P.C. Preston H. Smirman - Reg No. 36365
Signature	Preston H. Smirman
Date	December 18, 2003

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I hereby certify that this correspondence is being facsimile transmitted to the USPTO or deposited with the United States Postal Service with sufficient postage as first class mail in an envelope addressed to: Commissioner for Patents, P.O. Box 1450, Alexandria, VA 22313-1450 on the date shown below.

Typed or printed name	Preston H. Smirman - Reg No. 36365		
Signature	Preston H. Smirman	Date	December 18, 2003

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FEE TRANSMITTAL for FY 2004

Effective 10/01/2003. Patent fees are subject to annual revision.

☐ Applicant claims small entity status. See 37 CFR 1.27

TOTAL AMOUNT OF PAYMENT ($) 330.00

Complete if Known	
Application Number	10/016,472
Filing Date	December 10, 2001
First Named Inventor	Anthony J. Grzesiak et al.
Examiner Name	Melody M. Burch
Art Unit	3683
Attorney Docket No.	DKT 00065A (BWI-00056)

METHOD OF PAYMENT (check all that apply)

☑ Check ☐ Credit card ☐ Money Order ☐ Other ☐ None

☐ Deposit Account:

Deposit Account Number: 50-1612

Deposit Account Name: Warn, Burgess & Hoffmann

The Director is authorized to: (check all that apply)

☐ Charge fee(s) indicated below ☑ Credit any overpayments

☑ Charge any additional fee(s) or any underpayment of fee(s)

☐ Charge fee(s) indicated below, **except for the filing fee**

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FEE CALCULATION

1. BASIC FILING FEE

Large Entity Fee Code	Large Entity Fee ($)	Small Entity Fee Code	Small Entity Fee ($)	Fee Description	Fee Paid
1001	770	2001	385	Utility filing fee	
1002	340	2002	170	Design filing fee	
1003	530	2003	265	Plant filing fee	
1004	770	2004	385	Reissue filing fee	
1005	160	2005	80	Provisional filing fee	

SUBTOTAL (1) ($)

2. EXTRA CLAIM FEES FOR UTILITY AND REISSUE

			Extra Claims		Fee from below		Fee Paid
Total Claims		-20** =		X		=	
Independent Claims		- 3** =		X		=	
Multiple Dependent						=	

Large Entity Fee Code	Large Entity Fee ($)	Small Entity Fee Code	Small Entity Fee ($)	Fee Description
1202	18	2202	9	Claims in excess of 20
1201	86	2201	43	Independent claims in excess of 3
1203	290	2203	145	Multiple dependent claim, if not paid
1204	86	2204	43	** Reissue independent claims over original patent
1205	18	2205	9	** Reissue claims in excess of 20 and over original patent

SUBTOTAL (2) ($)

**or number previously paid, if greater; For Reissues, see above

FEE CALCULATION (continued)

3. ADDITIONAL FEES

Large Entity Fee Code	Large Entity Fee ($)	Small Entity Fee Code	Small Entity Fee ($)	Fee Description	Fee Paid
1051	130	2051	65	Surcharge - late filing fee or oath	
1052	50	2052	25	Surcharge - late provisional filing fee or cover sheet	
1053	130	1053	130	Non-English specification	
1812	2,520	1812	2,520	For filing a request for *ex parte* reexamination	
1804	920*	1804	920*	Requesting publication of SIR prior to Examiner action	
1805	1,840*	1805	1,840*	Requesting publication of SIR after Examiner action	
1251	110	2251	55	Extension for reply within first month	
1252	420	2252	210	Extension for reply within second month	
1253	950	2253	475	Extension for reply within third month	
1254	1,480	2254	740	Extension for reply within fourth month	
1255	2,010	2255	1,005	Extension for reply within fifth month	
1401	330	2401	165	Notice of Appeal	
1402	330	2402	165	Filing a brief in support of an appeal	330.00
1403	290	2403	145	Request for oral hearing	
1451	1,510	1451	1,510	Petition to institute a public use proceeding	
1452	110	2452	55	Petition to revive - unavoidable	
1453	1,330	2453	665	Petition to revive - unintentional	
1501	1,330	2501	665	Utility issue fee (or reissue)	
1502	480	2502	240	Design issue fee	
1503	640	2503	320	Plant issue fee	
1460	130	1460	130	Petitions to the Commissioner	
1807	50	1807	50	Processing fee under 37 CFR 1.17(q)	
1806	180	1806	180	Submission of Information Disclosure Stmt	
8021	40	8021	40	Recording each patent assignment per property (times number of properties)	
1809	770	2809	385	Filing a submission after final rejection (37 CFR 1.129(a))	
1810	770	2810	385	For each additional invention to be examined (37 CFR 1.129(b))	
1801	770	2801	385	Request for Continued Examination (RCE)	
1802	900	1802	900	Request for expedited examination of a design application	

Other fee (specify)

*Reduced by Basic Filing Fee Paid

SUBTOTAL (3) ($) 330.00

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SUBMITTED BY				(Complete (if applicable))	
Name (Print/Type)	Preston H. Smirman	Registration No. (Attorney/Agent)	36365	Telephone	(248) 364-4300
Signature	Preston H. Smirman			Date	December 18, 2003

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OIPE

DEC 2 4 2003

PATENT & TRADEMARK OFFICE

IN THE UNITED STATES PATENT AND TRADEMARK OFFICE
BEFORE THE BOARD OF PATENT APPEALS AND INTERFERENCES

In re application of:	Anthony J. Grzesiak et al.
Serial No.:	10/016,472
Filing Date:	December 10, 2001
Group Art Unit:	3683
Examiner:	Melody M. Burch
Title:	BRAKE BANDS FOR AN AUTOMATIC TRANSMISSION AND METHOD FOR CONTROLLING A GEAR SHIFT IN AUTOMATIC TRANSMISSION AND FEEDBACK LOOP CONTROL SYSTEM
Attorney Docket:	DKT 00065A (BWI-00056)

Certificate of Mailing

I hereby certify that this correspondence is being deposited with the United States Postal Service, as first class mail, in an envelope addressed to: Mail Stop Appeal Brief - Patents, Commissioner for Patents, P.O. Box 1450, Alexandria, VA 22313-1450, on December 18, 2003 By: Robert H. [signature]

12/30/2003 MDANTE1 00000018 10016472
01 FC:1402 330.00 OP

Mail Stop Appeal Brief - Patents
Commissioner for Patents
P.O. Box 1450
Alexandria, VA 22313-1450

Dear Sir:

Pursuant to 37 CFR §1.192, this is an Appeal Brief in response to the final Office Action mailed July 18, 2003. The Appeal Brief is submitted in triplicate.

A check in the amount of $330.00 is enclosed herewith for filing this Appeal Brief.

12/30/2003 MDANTE1 00000017 501612 10016472
01 FC:1402 330.00 DA

Void date: 12/30/2003 MDANTE1
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The Commissioner is hereby authorized to charge any fees that may be required for filing a brief in support of an appeal in accordance with 37 CFR §1.17(c), or credit any overpayment to Applicant's Deposit Account No. 501612.

If for some reason applicant has not requested a sufficient extension of time and/or has not paid a sufficient fee necessary to prevent abandonment of this application, please consider this as a Request for an Extension for the required time period and/or authorization to charge Applicants' Deposit Account No. 501612 for any extension of time fee which may be due.

Table of Cont nts

Table of Authoriti s

Real Party in Interest

The real party in interest for this appeal is BorgWarner, Inc. of Troy, Michigan, the assignee of the application.

Related Appeals and Interferences

There are no related appeals or interferences.

Status of the Claims

Claims 1, 4-9, 11-15 and 18-23 are pending in this application. Claims 2, 3, 10, 16, and 17 have been previously canceled, without prejudice.

Claims 1, 7 and 8 are rejected under 35 USC §102(b) as being anticipated by U.S. Patent No. 5,752,588 to Reichert et al. Claims 1, 7 and 8 are also rejected under 35 USC §102(b) as being anticipated by JP-11264460 (using U.S. Patent No. 6,102,825 to Hisano et al. as an English equivalent). Claims 4, 5, 9-11, 13, 18-21, and 23 are rejected under 35 USC §103(a) as being unpatentable over Reichert et al. in view of U.S. Patent No. 5,003,842 to Hatta et al. Claims 4, 5, 13-15 and 18-23 are rejected under 35 USC §103(a) as being unpatentable over JP-11264460 to Hisano et al. in view of U.S. Patent No. 5,003,842 to Hatta et al. Claim 6 is rejected under 35 USC §103(a) as being unpatentable over Reichert et al. in view of U.S. Patent No. 4,070,981 to Guinn et al. Claim 6 is rejected under 35 USC §103(a) as being unpatentable over JP-11264460 in view of U.S. Patent No. 4,070,981 to Guinn et al. Claim 12 is rejected under 35 USC §103(a) as being unpatentable over Reichert et al. in view of U.S. Patent No. 5,003,842 to Hatta et al. and further in view of Guinn et al.

In response to the amendment mailed April 28, 2003, claims 1, 4-9, 11-15 and 18-23 were finally rejected in the Office Action mailed July 18, 2003. A response to the final Office Action was mailed on September 18, 2003 and was considered and entered by the Examiner for purposes of appeal. An Advisory Action was mailed on October 14, 2003. This appeal is taken as to claims 1, 4-9, 11-15 and 18-23, as presently pending.

Status of Amendments

A response to the final Office Action was mailed on September 18, 2003 and was considered and entered by the Examiner for purposes of appeal. No further amendments have been submitted.

Summary of the Invention

A brake band mechanism (Page 4, line 19-Page 9, line 4 and Ref. 10 in Fig. 1 and Ref. 130 in Fig. 4) for an automatic transmission having a brake drum (Page 4, lines 5-9 and Ref. 12 in Fig. 1), said mechanism comprising a brake band (Page 4, line 5-Page 10, line 16 and Ref. 14 in Fig. 1) encircling the brake drum, said brake band including opposing ends (Page 5, line 6-Page 6, line 18 and Refs. 16 and 18 in Fig. 1), said brake band operable to be compressed and expanded around the brake drum (Page 5, line 7-Page 9, line 4) a two-stage hydraulic servo (Page 5, line 18-Page 10, line 16 and Ref. 28 in Figs. 1 and 3), and a linkage (Page 5, line 11-Page 7, line 2 and Refs. 22, 24, 26, 30, 32 and 34 in Fig. 1 and Refs. 102, 116, 118, 134, 138 and 140 in Fig.4) coupled to said servo and said brake band (Page 5, line 6-Page 9, line 4), said servo activating said linkage to provide positive compression and expansion to said brake band for applying friction to the brake drum to control the speed of rotation of said

brake drum (Page 5, line 18-Page 9, line 4), wherein said servo provides a rapid activation of said linkage during a first stage to rapidly expand said brake band, and a controlled compression and expansion of said brake band during a second stage (Page 5, line 18-Page 9, line 4), is claimed.

A brake band mechanism (Page 4, line 19-Page 9, line 4 and Ref. 10 in Fig. 1 and Ref. 130 in Fig. 4) for an automatic transmission having a brake drum (Page 4, lines 5-9 and Ref. 12 in Fig. 1), said mechanism comprising a brake band (Page 4, line 5-Page 10, line 16 and Ref. 14 in Fig. 1) encircling the brake drum, said brake band including opposing ends (Page 5, line 6-Page 6, line 18 and Refs. 16 and 18 in Fig. 1), said brake band operable to be compressed and expanded around the brake drum (Page 5, line 7-Page 9, line 4), a linkage coupled to said brake band (Page 5, line 11-Page 7, line 2 and Refs. 22, 24, 26, 30, 32 and 34 in Fig. 1 and Refs. 102, 116, 118, 134, 138 and 140 in Fig.4), a two-stage hydraulic servo (Page 5, line 18-Page 9, line 4 and Ref. 28 in Figs. 1 and 3), said linkage coupled to said servo (Page 5, line 6-Page 9, line 4), said servo including a servo rod position sensor (Page 8, lines 4-15 and Ref. 98 in Fig. 4) for determining a position of a stroke rod of said servo, said servo providing a rapid activation of the linkage during a first stage to rapidly expand said brake band, and a controlled compression and expansion of said brake band during a second stage (Page 5, line 18-Page 9, line 4), and a clip structure (Page 5, line 10-Page 7, line 2 and Ref. 21 and 36 in Fig. 1 and Ref. 132 and 136 in Fig. 4), said clip structure being mounted to an end of said brake band and being coupled to said linkage (Page 5, line 10-Page 7, line 2 and Ref. 21 and 36 in Fig. 1 and Ref. 132 and 136 in Fig. 4), said servo activating said linkage to provide positive compression and expansion to said brake band for applying friction to the brake drum to control the speed of rotation of said

brake drum (Page 5, line 18-Page 9, line 4), wherein said servo includes a first piston and a second piston (Page 7, line 3-Page 9, line 4 and Ref. 62 and 64 in Fig. 4), said first piston being smaller than said second piston (Page 7, line 3-Page 9, line 4), said first piston being operable to provide rapid movement of said brake band and said second piston being operable to provide fine adjustments of said brake band (Page 7, line 3-Page 9, line 4), is claimed.

A method of controlling a shift of an automatic transmission (Page 9, line 5-Page 10, line 22 and Fig. 7) comprising providing a brake band (Page 4, line 5-Page 10, line 16 and Ref. 14 in Fig. 1) for engaging a brake drum (Page 4, lines 5-9 and Ref. 12 in Fig. 1) of an automatic transmission, said brake band being positively controlled for both apply and release pressure around said brake drum (Page 5, line 7-Page 10, line 22), applying a first fast active compression force to said brake band to a predetermined position (Page 5, line 18-Page 10, line 22), and providing a closed loop control of pressure on said brake band in both positive apply and release directions for controlling shift parameters of the transmission based on a predetermined input (Page 9, line 5-Page 10, line 22), wherein a two-stage servo is used for controlling said brake band (Page 5, line 18-Page 10, line 22 and Ref. 28 in Figs. 1 and 3), wherein said servo has a first stage for rapidly applying band pressure, and a second stage for providing positive finite control of both apply and release pressures on said brake band during the shift (Page 5, line 18-Page 10, line 22), is claimed.

Summary of the References Cited

It is the Applicants' understanding that U.S. Patent No. 5,752,588 to Reichert et al. appears to disclose a hydraulic servo for friction brakes of an automatic transmission

that comprises a cylinder (2) with a main piston (3) arranged therein, a smaller cylinder (8) to receive a smaller compensation piston (9), the smaller cylinder communicating with the pressure chamber (5) of the main piston (3) through an orifice (18) controlled by a ball valve (20). A stepped bore (22) connects the pressure chamber (5) to the compensation chamber (17). A stepped piston (23), located in the bore, forms a discharge control valve (24), which controls the discharge of pressure medium from the compensation chamber (17).

It is the Applicants' understanding that JP-11264460 to Hisano et al. (using U.S. Patent No. 6,102,825 as an English language equivalent) appears to disclose a rotational direction of a rotational element at a high speed gear stage is different from a operational direction of a reaction torque at a low speed gear stage. The rotation by the reaction torque at the low speed gear stage is stopped with a self-energizing operation of a band brake operated by a hydraulic servo. A waiting pressure, which is lower for a predetermined amount than a engagement pressure at the self-energizing operation, and with which a basis of a racing amount occurs after a synchronizing point, is applied to the hydraulic servo until the synchronization is determined. The waiting pressure increases to the engagement pressure after determining the synchronization so that the rotational element is stopped gradually as preventing a long shift time.

It is the Applicants' understanding that U.S. Patent No. 5,003,842 to Hatta et al. appears to disclose, in an automatic transmission gear system which selectively actuates a plurality of frictional elements by oil pressure such as clutches and brakes that are provided in the transmission gear system in order to obtain different gear ratios, one frictional element is disengaged concurrently with engaging of another frictional element for switching the gear ratio (speed change). At that time, it is important to

adjust the timing for actuating the frictional elements depending on the conditions of the engine and the vehicle itself. The present invention relates to a control device for an automatic transmission gear system that assures smooth gear shifting at all times by adequately controlling the overlap of the torque capacities of the frictional elements.

It is the Applicants' understanding that U.S. Patent 4,070,981 to Guinn et al. appears to disclose a mooring system for maintaining a ship shape drilling vessel within alignment limits and for warping it into the sea while drilling a well from the vessel in the sub-surface ground below it. The mooring system absorbs all of the forces on the vessel, such as wind, current, wave, swell, roll, pitch, heave, surge and sway. These forces are measured by sensing load on a motor, electric, hydraulic, and the like, driving the anchor chain wildcat while hauling it in, by sensing load on the brake bands for the windlass wildcats, and by sensing load on the chain stopper, which sensed loads are transmitted to a display device which provides sufficient information to maintain the drilling vessel within the alignment limits and to warp it into the sea to minimize forces and motions of the vessel and to avoid beam sea forces. Preferably, a chain counter is provided on the power wildcat that counts the links, and hence the distance, the anchor chain is payed out or hauled in, which is transmitted to the display device. The anchor chains extend from each side both fore and aft of the vessel and each anchor chain has an electric motor driven wildcat, and extends through a chain stopper and fairlead. Vessel alignment is displayed on a cathode ray tube using an acoustic position resonance system.

Issues

Are Applicants' claims to a brake band mechanism for an automatic transmission having a brake drum, as recited in claims 1, 7 and 8, anticipated by U.S. Patent No. 5,752,588 to Reichert et al.?

Are Applicants' claims to a brake band mechanism for an automatic transmission having a brake drum, as recited in claims 1, 7 and 8, anticipated by JP-11264460 to Hisano et al.?

Are Applicants' claims to a brake band mechanism for an automatic transmission having a brake drum, and a method of controlling a shift of an automatic transmission, as recited in claims 4, 5, 9-11, 13, 18-21 and 23, unpatentable over U.S. Patent No. 5,752,588 to Reichert et al. in view of U.S. Patent No. 5,003,842 to Hatta et al.?

Are Applicants' claims to a brake band mechanism for an automatic transmission having a brake drum, and a method of controlling a shift of an automatic transmission, as recited in claims 4, 5, and 13-15, and 18-23, unpatentable over JP-11264460 to Hisano et al. in view of U.S. Patent No. 5,003,842 to Hatta et al.?

Are Applicants' claims to a brake band mechanism for an automatic transmission having a brake drum, as recited in claim 6, unpatentable over U.S. Patent No. 5,752,588 to Reichert et al. in view of U.S. Patent No. 4,070,981 to Guinn et al.?

Are Applicants' claims to a brake band mechanism for an automatic transmission having a brake drum, as recited in claim 6, unpatentable over JP-11264460 in view of U.S. Patent No. 4,070,981 to Guinn et al.?

Are Applicants' claims to a brake band mechanism for an automatic transmission having a brake drum, as recited in claim 12, unpatentable over U.S. Patent No. 5,003,842 to Hatta et al. as applied to claim 9, and further in view of Guinn et al.?

Grouping of the Claims

For purposes of this appeal, the claims stand and fall together.

Argument Regarding the 35 USC §102(b) Rejection of Claims 1, 7 and 8

Claims 1, 7 and 8 stand rejected under 35 U.S.C. §102(b), as being anticipated by U.S. Patent No. 5,752,588 to Reichert et al.

The Applicants respectfully traverse the 35 U.S.C. §102(b) rejection of claims 1, 7 and 8.

The law is clear that anticipation requires that a single prior art reference disclose each and every limitation of the claim sought to be rejected. 35 U.S.C. §102(b).

The law is also clear that a claim in dependent form shall be construed to incorporate all the limitations of the claim to which it refers. 35 U.S.C. §112, fourth paragraph.

With respect to the recitation of claim 1, as amended, the Applicants submit that Reichert et al. fails in teaching the claimed structure.

While Reichert et al. may arguably disclose a two-stage hydraulic circuit, there is no teaching that the "servo provides a rapid activation of [the] linkage during a first stage to rapidly expand [the] brake band, and a controlled compression and expansion of [the] brake band during a second stage."

With respect to the Examiner's assertion that "it is evident that Reichert et al. describe the invention to the same extent as Applicant," the Applicants respectfully disagree.

There is no teaching whatsoever in Reichert et al. regarding rapid activation of a linkage during a first stage to rapidly expand the brake band, and a controlled

compression and expansion of the brake band during a second stage. Reichert et al. is only concerned with hydraulic fluid conservation, not rapid and/or controlled linkage actuation, as presently claimed.

With respect to the Examiner's assertions that "the use of a minimized amount of hydraulic fluid to achieve actuation suggests that actuation occurs faster since it takes less time to wait for the accumulation of a small or minimized volume of fluid" and "since the smaller apply piston associated with the first stage is the first to cause brake actuation, the first stage may be considered the quicker (or comparatively rapid) stage just as the first runner to reach a finish line of a race is considered to be the quicker runner," the Applicants respectfully disagree.

While Reichert et al. may arguably disclose a two-stage hydraulic circuit, there is no teaching that the servo provides a rapid activation of the linkage during a first stage to rapidly expand the brake band, and a controlled compression and expansion of the brake band during a second stage.

Conversely, Reichert et al. discloses, at column 1, lines 33-39, that:

> It is an object of the invention to provide an hydraulic servo with travel compensation, for friction brakes for shifting an automatic transmission for motor vehicles, in order, at the time of shift, **to minimize the volume of hydraulic fluid required to apply a friction brake** to avoid an undesired pressure drop due to the volume of fluid which has to be made available. (Emphasis added).

Thus, Reichert et al. appears to disclose that the supposed first stage actuation of the hydraulic servo is accomplished slowly, due to the conservation of hydraulic fluid delivered to the pressure chamber of the supposed main piston of the servo. More specifically, Reichert et al. is concerned primarily with conserving hydraulic fluid in the event of a system leak, e.g., through the use of a compensation pressure chamber and

cooperating piston, than with rapid first stage piston actuation of the hydraulic servo, as presently claimed.

Accordingly, the Applicants contend that the 35 U.S.C. §102(b) rejection of claims 1, 7 and 8 has been overcome.

Argument Regarding the 35 USC §102(b) Rejection of Claims 1, 7 and 8

Claims 1, 7 and 8 stand rejected under 35 U.S.C. §102(b), as being anticipated by JP-11264460 (using U.S. Patent No. 6,102,825 to Hisano et al. as an English equivalent).

The Applicants respectfully traverse the 35 U.S.C. §102(b) rejection of claims 1, 7 and 8.

Hisano et al. teaches no such structure as recited in claim 1, as amended.

Specifically, while Hisano et al. may arguably disclose a two-stage hydraulic circuit, there is no teaching that the "servo provides a rapid activation of [the] linkage during a first stage to rapidly expand [the] brake band, and a controlled compression and expansion of [the] brake band during a second stage."

With respect to the Examiner's assertion that "since the Hisano et al. reference shows a small apply piston 43 arranged closest to the linkage that promotes the initial brake application movement to the linkage and shows a larger apply piston 44 for more finite adjustments of the brake band pressure to the same extent as Applicant's, Examiner maintains the rejections," the Applicants respectfully disagree.

While Hisano et al. may arguably disclose a two-stage hydraulic circuit, there is no teaching or suggestion that the "servo provides a rapid activation of [the] linkage during a first stage to rapidly expand [the] brake band, and a controlled compression

and expansion of [the] brake band during a second stage." For example, Hisano et al. disclose at, column 3, lines 3-23:

> According to the invention, the rotation of the rotational element reduces to synchronize with the rotation of the rotational element at the low speed gear stage. That is, the rotation of the rotational element reduces to stop. In this case, the de-energizing operation occurs at the band brake. Therefore, **<u>the rotational element is not stopped from rotating by the band brake, because the engagement force occurred by the application of the aforementioned hydraulic pressure is small</u>**. After that, when the rotational element is stopped from rotating and then the reverse rotation of the rotational element is started, the self-energizing operation occurs. Therefore, the engagement force of the band brake steeply increases to stop the rotational element from rotating.
>
> In this case, the hydraulic pressure applied to the hydraulic servo of the band brake is the waiting pressure, which is lower for the predetermined amount than the hydraulic pressure to maintain the stop of the rotation of the rotational element. Therefore, **<u>the rotational element is not steeply stopped, that is, the rotation of the rotational element changes gradually</u>**. (Emphasis added).

Thus, Hisano et al. appears to disclose that the supposed first stage actuation of the hydraulic servo is accomplished slowly or weakly, due to the application of only a small amount of hydraulic pressure to the supposed main piston of the servo.

Accordingly, the Applicants contend that the 35 U.S.C. §102(b) rejection of 1, 7 and 8 has been overcome.

<u>Argument Regarding the 35 USC §103(a) Rejection of Claims 4, 5, 9-11, 13, 18-21 and 23</u>

Claims 4, 5, 9-11, 13, 18-21, and 23 stand rejected under 35 U.S.C. §103(a), as being unpatentable over Reichert et al. in view of U.S. Patent No. 5,003,842 to Hatta et al.

The Applicant respectfully traverses the 35 U.S.C. §103(a) rejection of claims 4, 5, 9-11, 13, 16-21, and 23. The Examiner should note that claim 10 was canceled, without prejudice, in a previous response.

The standard for obviousness is that there must be some suggestion, either in the reference or in the relevant art, of how to modify what is disclosed to arrive at the claimed invention. In addition, "[s]omething in the prior art as a whole must suggest the desirability and, thus, the obviousness, of making" the modification to the art suggested by the Examiner. *Uniroyal, Inc. v. Rudkin-Wiley Corp.*, 837 F.2d 1044, 1051, 5 U.S.P.Q.2d (BNA) 1434, 1438 (Fed. Cir.), cert. denied, 488 U.S. 825 (1988). Although the Examiner may suggest the teachings of a primary reference could be modified to arrive at the claimed subject matter, the modification is not obvious unless the prior art also suggests the desirability of such modification. *In re Laskowski*, 871 F.2d 115, 117, 10 U.S.P.Q.2d (BNA) 1397, 1398 (Fed. Cir.1989). There must be a teaching in the prior art for the proposed combination or modification to be proper. *In re Newell*, 891 F.2d 899, 13 U.S.P.Q.2d (BNA) 1248 (Fed. Cir. 1989). If the prior art fails to provide this necessary teaching, suggestion, or incentive supporting the Examiner's suggested modification, the rejection based upon this suggested modification is error and must be reversed. *In re Bond*, 910 F.2d 831, 15 U.S.P.Q.2d (BNA) 1566 (Fed. Cir. 1990).

The Examiner apparently cited Hatta et al. in order to cure the aforementioned deficiencies in the disclosure of Reichert et al. Although Hatta et al. may arguably suggest a piston position sensor, the Examiner is correct that Hatta et al. fail to teach or suggest a servo rod position sensor. However, Hatta et al. adds nothing to the disclosure of Reichert et al. in terms of disclosing the structure and function of the two-stage hydraulic servo, wherein the servo provides a rapid activation of the linkage

during a first stage to rapidly expand the brake band, and a controlled compression and expansion of the brake band during a second stage, as presently recited in claim 1, as amended.

Because claim 1 is allowable over Reichert et al. and/or Hatta et al., either alone or in combination therewith, for at least the reasons stated above, claims 4 and 5, which further define claim 1, are likewise allowable.

Accordingly, the 35 USC §103(a) rejection of claims 4 and 5 has been overcome.

Neither Reichert et al. and/or Hatta et al., either alone or in combination therewith, teach or suggest such a structure as recited in claim 9, as amended.

Specifically, while Reichert et al. may arguably disclose a two-stage hydraulic circuit, there is no teaching of a "servo providing a rapid activation of the linkage during a first stage to rapidly expand said brake band, and a controlled compression and expansion of said brake band during a second stage; and ... [the] servo activating said linkage to provide positive compression and expansion to said brake band for applying friction to the brake drum to control the brake drum's speed of rotation; wherein said servo includes a first piston and a second piston, said first piston being smaller than said second piston, said first piston being operable to provide rapid movement of said brake band and said second piston being operable to provide fine adjustments of said brake band."

As previously noted, although Hatta et al. may arguably suggest a piston position sensor, the Examiner is correct that Hatta et al. fail to teach or suggest a servo rod position sensor. However, Hatta et al. does not cure the aforementioned deficiencies in the disclosure of Reichert et al.

Accordingly, the 35 U.S.C. §103(a) rejection of claim 9 has been overcome.

Because claim 9 is allowable over Reichert et al. and/or Hatta et al., either alone or in combination therewith, for at least the reasons stated above, claim 11, which further defines claim 9, is likewise allowable.

Neither Reichert et al. and/or Hatta et al., either alone or in combination therewith, teach or suggest such methodology as recited in claim 13, as amended.

Specifically, while Reichert et al. may arguably disclose a two-stage hydraulic circuit, there is no teaching of a method for "applying a first fast active compression force to said brake band to a predetermined position ... wherein a two-stage servo is used for controlling said brake band; wherein said servo has a first stage for rapidly applying band pressure, and a second stage for providing positive finite control of both apply and release pressures on said brake band during the shift."

Again, as previously noted, although Hatta et al. may arguably suggest a piston position sensor, the Examiner is correct that Hatta et al. fail to teach or suggest a servo rod position sensor. However, Hatta et al. does not cure the aforementioned deficiencies in the disclosure of Reichert et al.

Accordingly, the 35 U.S.C. §103(a) rejection of claim 13 has been overcome.

Because claim 13 is allowable over Reichert et al. and/or Hatta et al., either alone or in combination therewith, for at least the reasons stated above, claims 18-21 and 23, which further define claim 13, are likewise allowable.

<u>Argument Regarding the 35 USC §103(a) Rejection of Claims 4, 5, 13-15 and 18-23</u>

Claims 4, 5, and 13-15, and 18-23 are rejected under 35 USC §103(a) as being unpatentable over JP-11264460 to Hisano et al. in view of U.S. Patent No. 5,003,842 to Hatta et al.

The Applicants respectfully traverse the 35 U.S.C. §103(a) rejection of claims 4, 5, 13-15, and 18-23.

The Examiner apparently cited Hatta et al. in order to cure the aforementioned deficiencies in the disclosure of Hisano et al. Again, as previously noted, although Hatta et al. may arguably suggest a piston position sensor, the Examiner is correct that Hatta et al. fail to teach or suggest a servo rod position sensor. However, Hatta et al. adds nothing to the disclosure of Hisano et al. in terms of disclosing the structure and function of the two stage hydraulic servo of the invention, as presently recited in claim 1, as amended.

Because claim 1 is allowable over Hisano et al. and/or Hatta et al., either alone or in combination therewith, for at least the reasons stated above, claims 4 and 5, which further define claim 1, are likewise allowable.

Accordingly, the 35 USC §103(a) rejection of claims 4 and 5 has been overcome.

Furthermore, neither Hisano et al. and/or Hatta et al., either alone or in combination therewith, teach or suggest such methodology as claimed in claim 13.

Specifically, while Hisano et al. may arguably disclose a two-stage hydraulic circuit, there is no teaching of a method for "applying a first fast active compression force to said brake band to a predetermined position ... wherein a two-stage servo is used for controlling said brake band; wherein said servo has a first stage for rapidly applying band pressure, and a second stage for providing positive finite control of both apply and release pressures on said brake band during the shift."

Again, as previously noted, although Hatta et al. may arguably suggest a piston position sensor, the Examiner is correct that Hatta et al. fail to teach or suggest a servo

rod position sensor. However, Hatta et al. does not cure the aforementioned deficiencies in the disclosure of Hisano et al.

Accordingly, the 35 USC §103(a) rejection of claim 13 has been overcome.

Because claim 13 is allowable over Hisano et al. and/or Hatta et al., either alone or in combination therewith, for at least the reasons stated above, claims 14, 15 and 18-23, which further define claim 13, are likewise allowable.

Argument Regarding the 35 USC §103(a) Rejection of Claim 6

Claim 6 is rejected under 35 USC §103(a) as being unpatentable over Reichert et al. in view of U.S. Patent No. 4,070,981 to Guinn et al.

The Applicants respectfully traverse the 35 USC §103(a) rejection of claim 6.

The Examiner apparently cited Guinn et al. in order to cure the aforementioned deficiencies in the disclosure of Reichert et al. Although Guinn et al. may arguably disclose a strain sensor, it adds nothing to the disclosure of Reichert et al. in terms of disclosing the structure and function of the two-stage hydraulic servo, wherein the servo provides a rapid activation of the linkage during a first stage to rapidly expand the brake band, and a controlled compression and expansion of the brake band during a second stage, as presently recited in claim 1, as amended.

Furthermore, Guinn et al. discloses a mooring system for floating drilling vessels and does not appear to even mention automatic transmissions. Thus, one of ordinary skill in the art would not look to Guinn et al. for guidance on constructing or operating an automatic transmission as presently claimed.

Because claim 1 is allowable over Reichert et al. for at least the reasons stated above, claim 6, which further defines claim 1, is likewise allowable.

Accordingly, the 35 USC §103(a) rejection of claim 6 has been overcome.

Argument Regarding the 35 USC §103(a) Rejection of Claim 6

Claim 6 is rejected under 35 USC §103(a) as being unpatentable over JP-11264460 in view of U.S. Patent No. 4,070,981 to Guinn et al.

The Applicants respectfully traverse the 35 USC §103(a) rejection of claim 6.

The Examiner apparently cited Guinn et al. in order to cure the aforementioned deficiencies in the disclosure of Hisano et al. Although Guinn et al. may arguably disclose a strain sensor, it adds nothing to the disclosure of Hisano et al. in terms of disclosing the structure and function of the two-stage hydraulic servo, wherein the servo provides a rapid activation of the linkage during a first stage to rapidly expand the brake band, and a controlled compression and expansion of the brake band during a second stage, as presently recited in claim 1, as amended.

Furthermore, Guinn et al. discloses a mooring system for floating drilling vessels and does not appear to even mention automatic transmissions. Thus, one of ordinary skill in the art would not look to Guinn et al. for guidance on constructing or operating an automatic transmission as presently claimed.

Because claim 1 is allowable over Hisano et al. for at least the reasons stated above, claim 6, which further defines claim 1, is likewise allowable.

Accordingly, the 35 USC §103(a) rejection of claim 6 has been overcome.

Argument Regarding the 35 USC §103(a) Rejection of Claim 12

Claim 12 is rejected under 35 USC §103(a) as being unpatentable over Reichert et al. in view of U.S. Patent No. 5,003,842 to Hatta et al. as applied to claim 9 above, and further in view of Guinn et al.

The Applicants respectfully traverse the 35 USC §103(a) rejection of claim 12.

The Examiner apparently cited Guinn et al. in order to cure the aforementioned deficiencies in the disclosure of Reichert et al. and/or Hatta et al. Although Guinn et al. may arguably disclose a strain sensor, it adds nothing to the disclosure of Reichert et al. and/or Hatta et al. in terms of disclosing the structure and function of the two-stage hydraulic servo, wherein the servo provides a rapid activation of the linkage during a first stage to rapidly expand the brake band, and a controlled compression and expansion of the brake band during a second stage, as presently recited in claim 9, as amended.

Furthermore, Guinn et al. discloses a mooring system for floating drilling vessels and does not appear to even mention automatic transmissions. Thus, one of ordinary skill in the art would not look to Guinn et al. for guidance on constructing or operating an automatic transmission as presently claimed.

Because claim 9 is allowable over Reichert et al. and/or Hatta et al., either alone or in combination therewith, for at least the reasons stated above, claim 12, which further defines claim 9, is likewise allowable.

Accordingly, the 35 USC §103(a) rejection of claim 12 has been overcome.

Conclusion

For the reasons advanced above, appellant respectfully urges that the rejections of claims 1, 4-9, 11-15 and 18-23 under 35 USC §§102(b) and/or 103(a) are improper. Reversal of the rejections in this appeal is respectfully requested.

Respectfully submitted,

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APPENDIX

COPY OF CLAIMS INVOLVED IN THE APPEAL

1. A brake band mechanism for an automatic transmission having a brake drum, said mechanism comprising:

a brake band encircling the brake drum, said brake band including opposing ends, said brake band operable to be compressed and expanded around the brake drum;

a two-stage hydraulic servo; and

a linkage coupled to said servo and said brake band, said servo activating said linkage to provide positive compression and expansion to said brake band for applying friction to the brake drum to control the speed of rotation of said brake drum;

wherein said servo provides a rapid activation of said linkage during a first stage to rapidly expand said brake band, and a controlled compression and expansion of said brake band during a second stage.

4. The mechanism according to claim 1 further comprising a position sensor, said position sensor sensing the position of a piston of said servo.

5. The mechanism according to claim 1 further comprising at least one linkage sensor, said at least one linkage sensor sensing the position of said linkage.

6. The mechanism according to claim 1 further comprising at least one band strain sensor, said at least one band strain sensor measuring the strain on said brake band.

7. The mechanism according to claim 1 wherein said servo includes a first piston and a second piston, said first piston being smaller than said second piston, said first piston being operable to provide rapid movement of said brake band and said second piston being operable to provide fine adjustments of said brake band.

8. The mechanism according to claim 1 further comprising a clip structure, said clip structure being mounted to at least one of the opposing ends of said brake band and being coupled to said linkage.

9. A brake band mechanism for an automatic transmission having a brake drum, said mechanism comprising:

a brake band encircling the brake drum, said brake band including opposing ends, said brake band operable to be compressed and expanded around the brake drum;

a linkage coupled to said brake band;

a two-stage hydraulic servo, said linkage coupled to said servo, said servo including a servo rod position sensor for determining a position of a stroke rod of said servo, said servo providing a rapid activation of the linkage during a first stage to rapidly expand said brake band, and a controlled compression and expansion of said brake band during a second stage; and

a clip structure, said clip structure being mounted to an end of said brake band and being coupled to said linkage, said servo activating said linkage to provide positive compression and expansion to said brake band for applying friction to the brake drum to control the speed of rotation of said brake drum;

wherein said servo includes a first piston and a second piston, said first piston being smaller than said second piston, said first piston being operable to provide rapid movement of said brake band and said second piston being operable to provide fine adjustments of said brake band.

11. The mechanism according to claim 9 further comprising at least one linkage sensor, said at least one linkage sensor sensing the position of said linkage.

12. The mechanism according to claim 9 further comprising at least one band strain sensor, said at least one band strain sensor measuring the strain on said brake band.

13. A method of controlling a shift of an automatic transmission comprising:

providing a brake band for engaging a brake drum of an automatic transmission, said brake band being positively controlled for both apply and release pressure around said brake drum;

applying a first fast active compression force to said brake band to a predetermined position; and

providing a closed loop control of pressure on said brake band in both positive apply and release directions for controlling shift parameters of the transmission, based on a predetermined input;

wherein a two-stage servo is used for controlling said brake band;

wherein said servo has a first stage for rapidly applying band pressure, and a second stage for providing positive finite control of both apply and release pressures on said brake band during the shift.

14. The method of claim 13 wherein said shift parameters are selected from the group consisting of servo position, apply strut strain, servo pressure, band strain, engine RPM, transmission torque output, and combinations thereof.

15. The method of claim 13 further comprising a closed loop software control system controlling an apply solenoid.

18. The method of claim 13 wherein said method comprises controlling said shift by first ramping up the pressure at the beginning of said shift and releasing pressure toward the end of said shift.

19. The method of claim 18 wherein said brake band is locked in an applied position after the completion of said shift.

20. The method of claim 18 wherein a switch between ramping up and closed loop control is determined by inputs selected from the group consisting of servo position, apply strut strain, servo pressure, band strain, engine RPM, transmission torque output, and combinations thereof.

21. The method of claim 18 wherein both apply and release pressures are independently controlled.

22. The method of claim 21 wherein solenoids are used to independently control the apply and release hydraulic pressure.

23. The method of claim 13 wherein said first stage is a smaller volume piston than said second stage.

US006102825A

United States Patent [19]

Hisano et al.

[11] **Patent Number: 6,102,825**

[45] **Date f Patent: Aug. 15, 2000**

[54] **CONTROL SYSTEM FOR AUTOMATIC TRANSMISSION**

[75] Inventors: **Takayuki Hisano**; **Masahiro Hayabuchi**; **Satoru Kasuya**, all of Anjo; **Tatsuya Iida**, Takefu, all of Japan

[73] Assignee: **Aisin AW Co., Ltd.**, Anjo, Japan

[21] Appl. No.: **09/268,709**

[22] Filed: **Mar. 16, 1999**

[30] **Foreign Application Priority Data**

Mar. 17, 1998 [JP] Japan 10-089286

[51] **Int. Cl.**^{7} **F16H 61/06**

[52] **U.S. Cl.** **475/118**; 475/120; 475/127

[58] **Field of Search** 475/116, 118, 475/120, 127

[56] **References Cited**

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Primary Examiner—Sherry L. Estremsky
Attorney, Agent, or Firm—Oliff & Berridge, PLC

[57] **ABSTRACT**

A rotational direction of a rotational element at a high speed gear stage is different from a operational direction of a reaction torque at a low speed gear stage. The rotation by the reaction torque at the low speed gear stage is stopped with a self-energizing operation of a band brake operated by a hydraulic servo. A waiting pressure, which is lower for a predetermined amount than a engagement pressure at the self-energizing operation, and with which a basis of a racing amount occurs after a synchronizing point, is applied to the hydraulic servo until the synchronization is determined. The waiting pressure increases to the engagement pressure after determining the synchronization so that the rotational element is stopped gradually as preventing a long shift time.

16 Claims, 13 Drawing Sheets

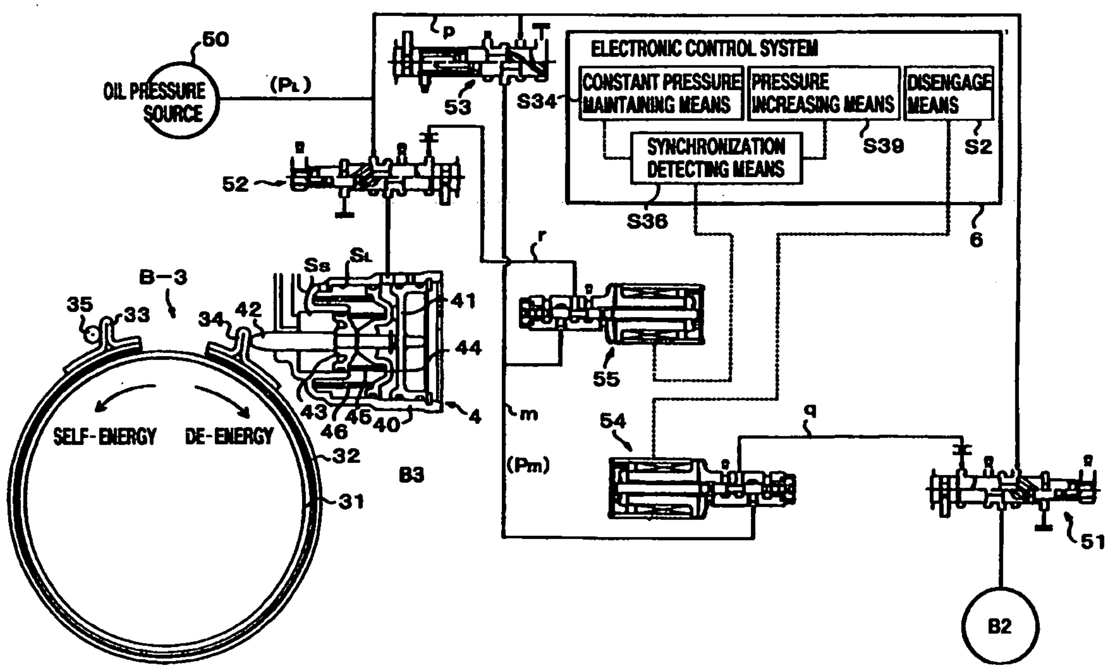
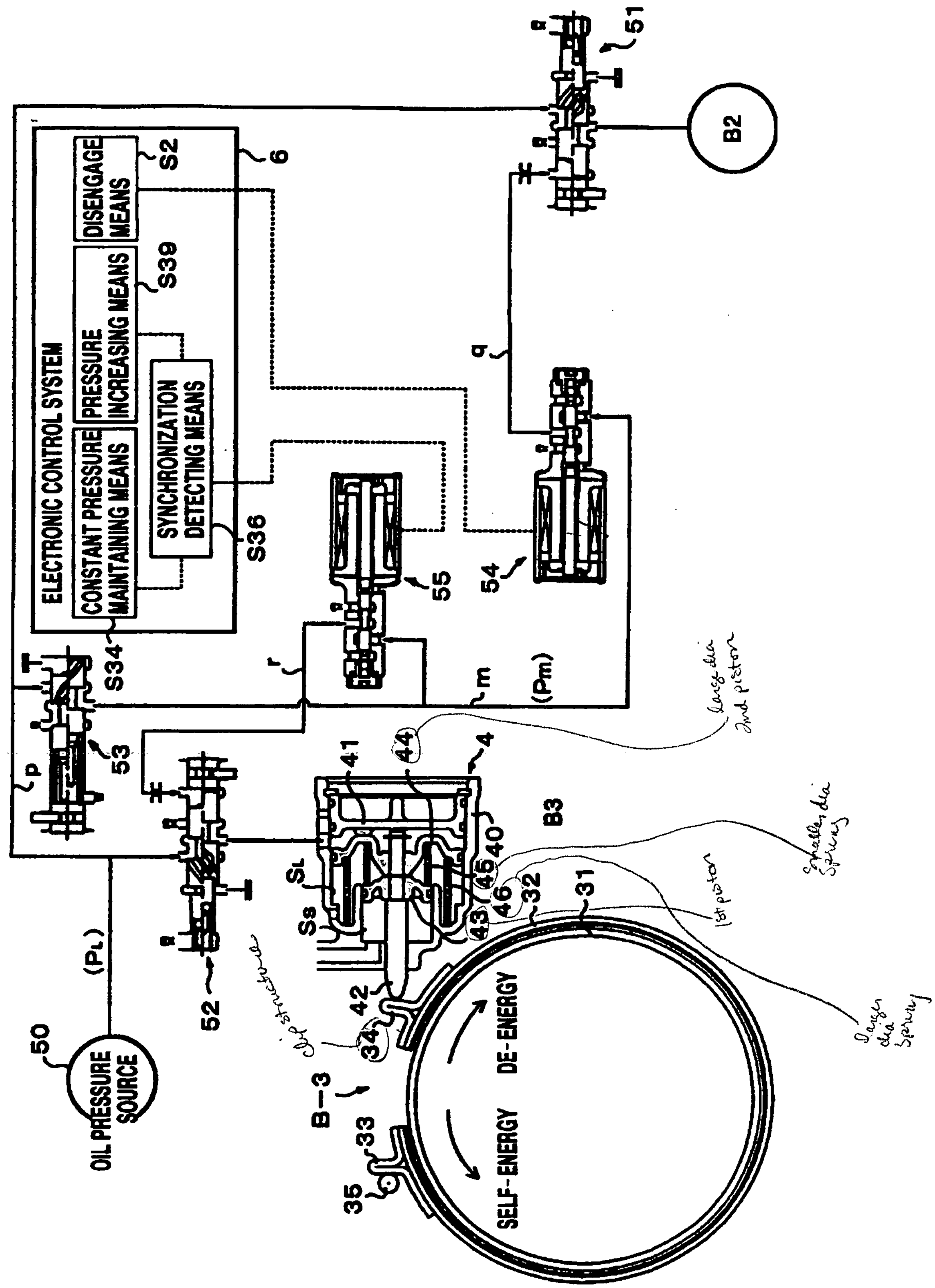
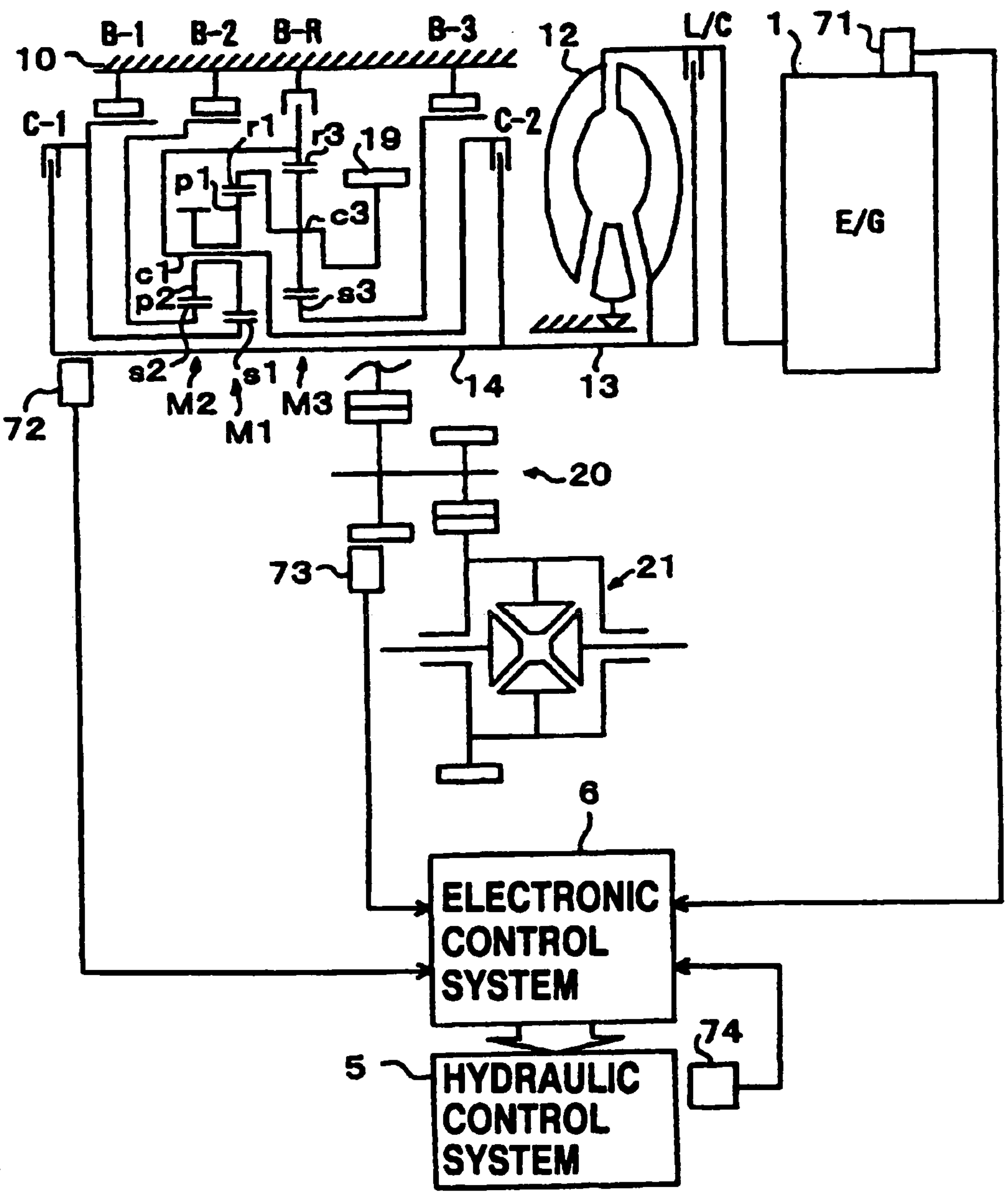


FIG. 1



ELECTRONIC CONTROL SYSTEM
CONSTANT PRESSURE MAINTAINING MEANS
PRESSURE INCREASING MEANS
DISENGAGE MEANS
SYNCHRONIZATION DETECTING MEANS
S34
S36
S39
S2
6
51
B2
q
54
55
r
m
(Pm)
53
p
52
(PL)
OIL PRESSURE SOURCE
50
4
41
44
40
45
46
43
32
31
B3
SL
Ss
42
34
33
35
B-3
DE-ENERGY
SELF-ENERGY
large dia 2nd piston
smaller dia spring
1st piston
larger dia spring
clip structure

FIG. 2



B-1
B-2
B-R
B-3
12
L/C
1
71
10
C-1
r1
r3
19
C-2
p1
c3
E/G
c1
p2
s3
s2
s1
M2
M1
M3
14
13
72
20
73
21
6
ELECTRONIC CONTROL SYSTEM
5
HYDRAULIC CONTROL SYSTEM
74

FIG. 3

	C-1	C-2	B-1	B-2	B-3	B-R
P	O					
REV	O					O
N	O					
1ST	O				O	
2ND		O			O	
3RD	O	O				
4TH		O	O			
5TH		O		O		

FIG. 4

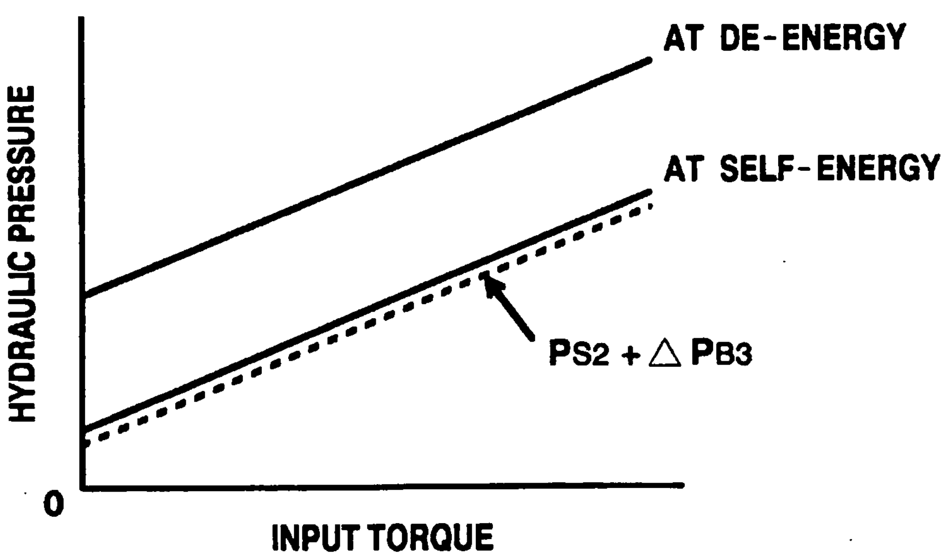


FIG. 5

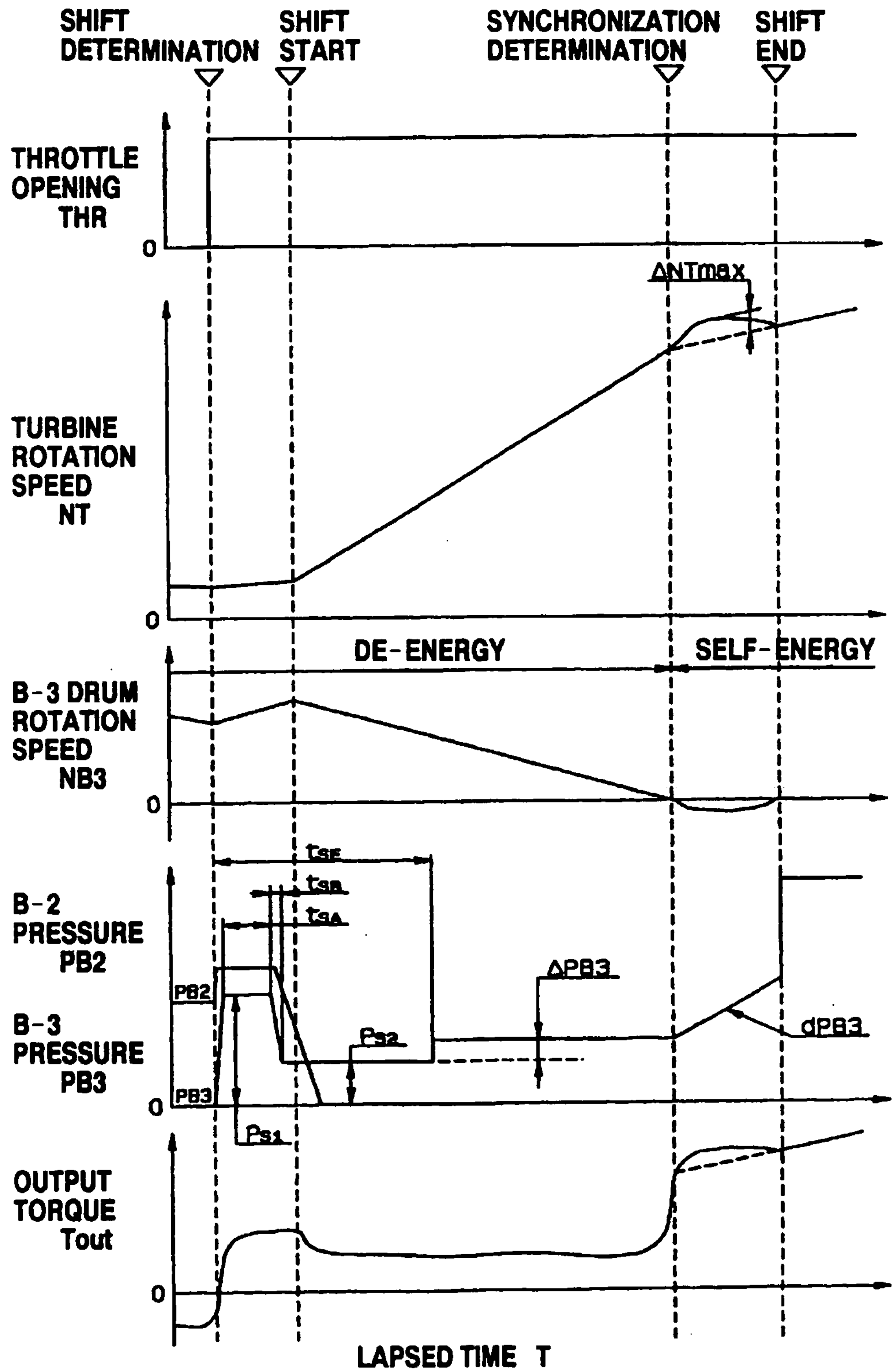

SHIFT DETERMINATION
SHIFT START
SYNCHRONIZATION DETERMINATION
SHIFT END
THROTTLE OPENING THR
0
ΔNTmax
TURBINE ROTATION SPEED NT
0
DE-ENERGY
SELF-ENERGY
B-3 DRUM ROTATION SPEED NB3
0
tSE
tSB
tSA
B-2 PRESSURE PB2
PB2
ΔPB3
Ps2
dPB3
B-3 PRESSURE PB3
PB3
0
Ps1
OUTPUT TORQUE Tout
0
LAPSED TIME T

FIG. 6

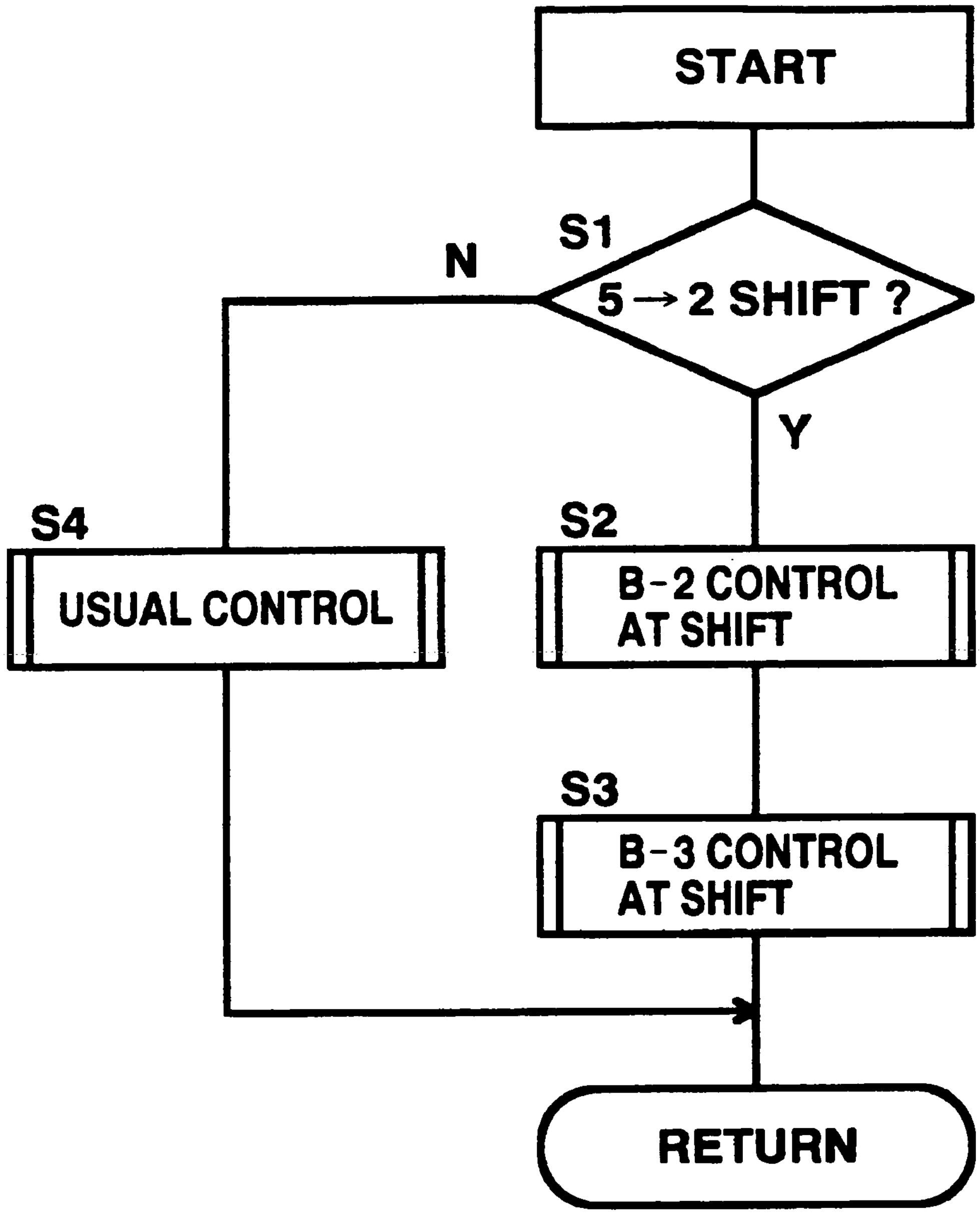

START
S1
N
5→2 SHIFT ?
Y
S4
USUAL CONTROL
S2
B-2 CONTROL AT SHIFT
S3
B-3 CONTROL AT SHIFT
RETURN

FIG. 7

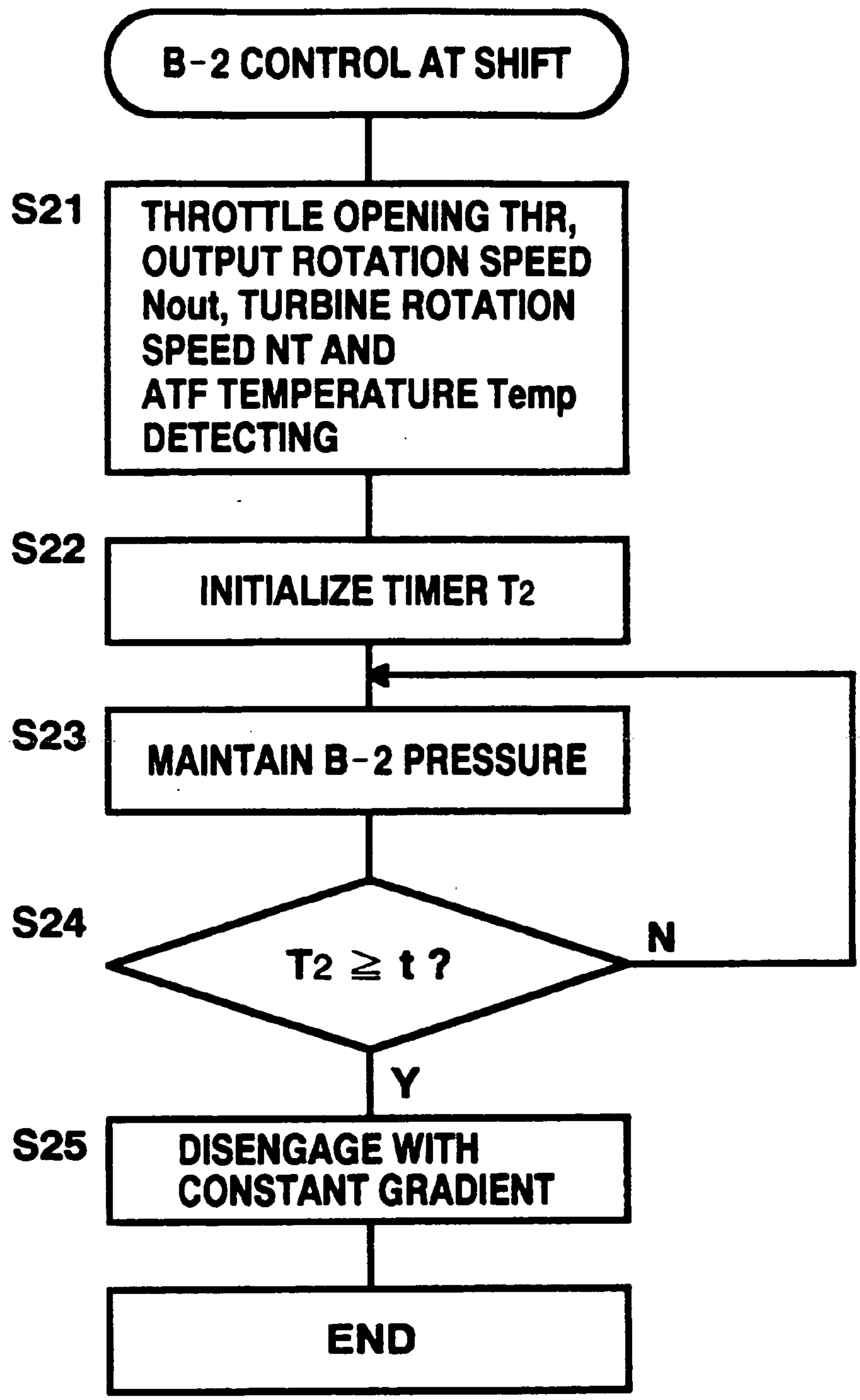

B-2 CONTROL AT SHIFT
S21
THROTTLE OPENING THR, OUTPUT ROTATION SPEED Nout, TURBINE ROTATION SPEED NT AND ATF TEMPERATURE Temp DETECTING
S22
INITIALIZE TIMER T2
S23
MAINTAIN B-2 PRESSURE
S24
T2 ≧ t ?
N
Y
S25
DISENGAGE WITH CONSTANT GRADIENT
END

FIG. 8

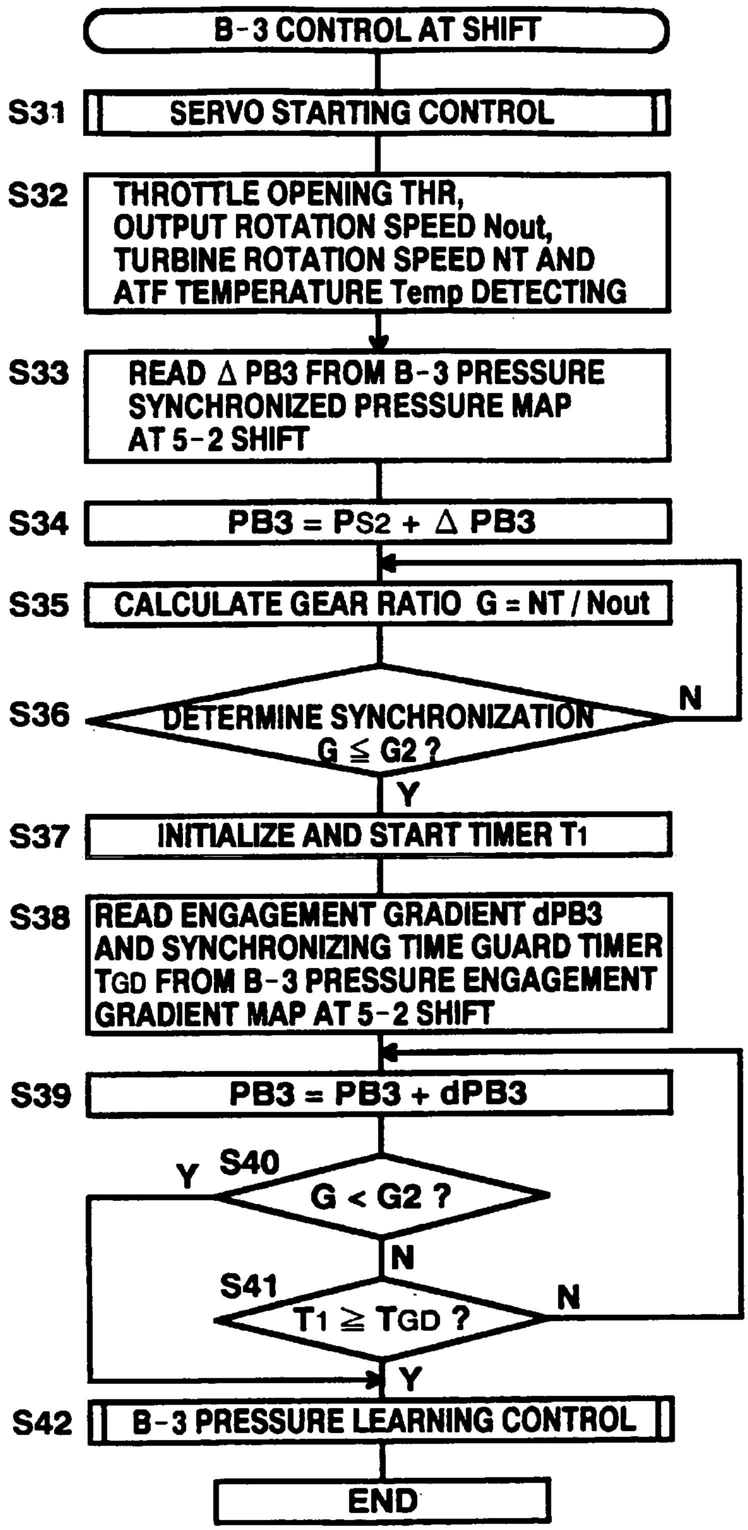

B-3 CONTROL AT SHIFT
S31
SERVO STARTING CONTROL
S32
THROTTLE OPENING THR, OUTPUT ROTATION SPEED Nout, TURBINE ROTATION SPEED NT AND ATF TEMPERATURE Temp DETECTING
S33
READ Δ PB3 FROM B-3 PRESSURE SYNCHRONIZED PRESSURE MAP AT 5-2 SHIFT
S34
PB3 = PS2 + Δ PB3
S35
CALCULATE GEAR RATIO G = NT / Nout
S36
DETERMINE SYNCHRONIZATION G ≦ G2 ?
N
Y
S37
INITIALIZE AND START TIMER T1
S38
READ ENGAGEMENT GRADIENT dPB3 AND SYNCHRONIZING TIME GUARD TIMER TGD FROM B-3 PRESSURE ENGAGEMENT GRADIENT MAP AT 5-2 SHIFT
S39
PB3 = PB3 + dPB3
S40
Y
G < G2 ?
N
S41
T1 ≧ TGD ?
N
Y
S42
B-3 PRESSURE LEARNING CONTROL
END

FIG. 9

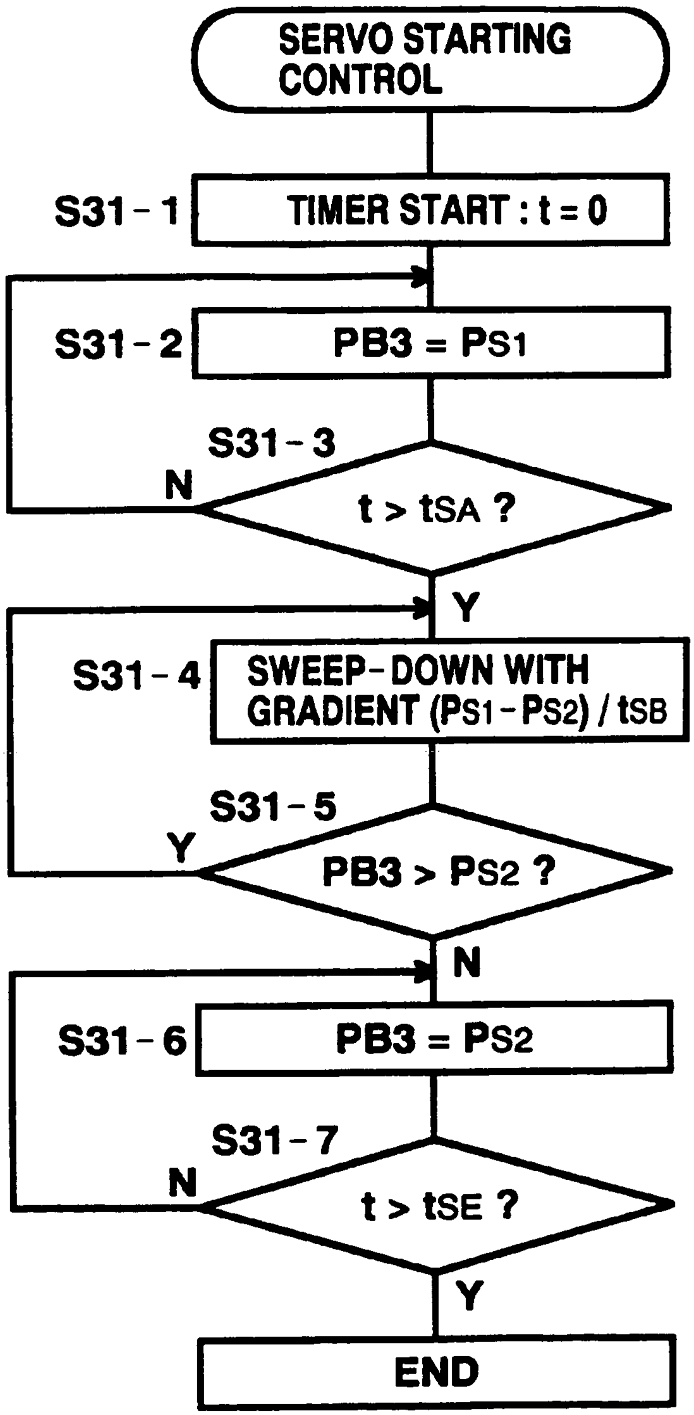

SERVO STARTING CONTROL
S31-1
TIMER START : t = 0
S31-2
PB3 = PS1
S31-3
N
t > tSA ?
Y
S31-4
SWEEP-DOWN WITH GRADIENT (PS1-PS2) / tSB
S31-5
Y
PB3 > PS2 ?
N
S31-6
PB3 = PS2
S31-7
N
t > tSE ?
Y
END

FIG. 10

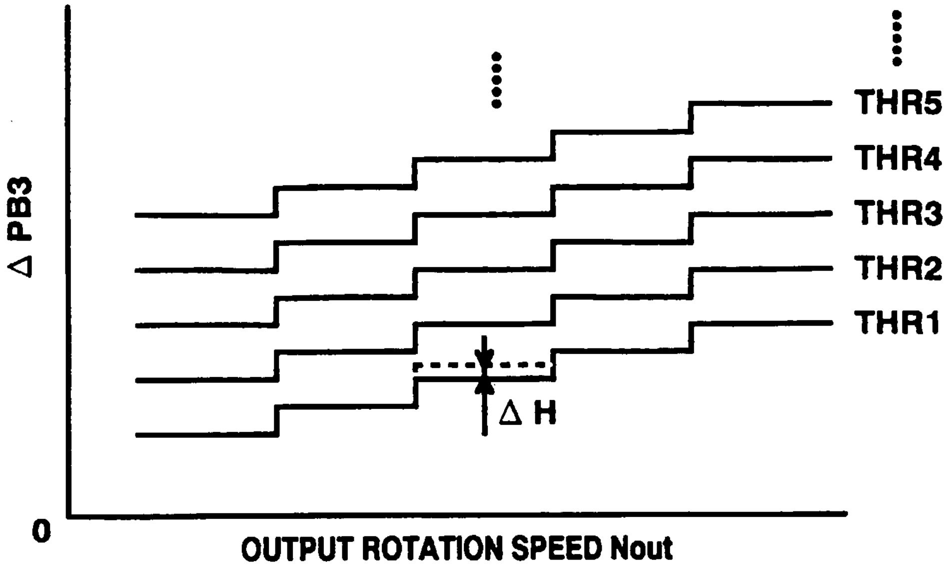


FIG. 11

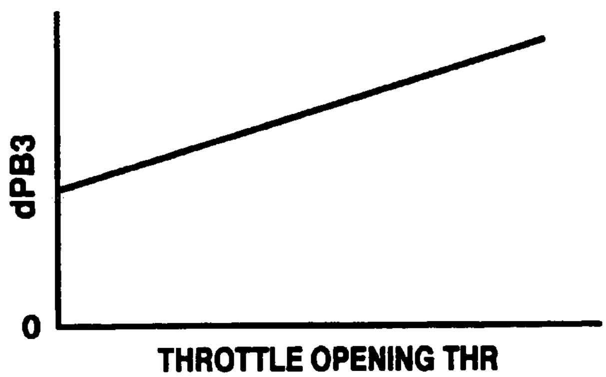


FIG. 12

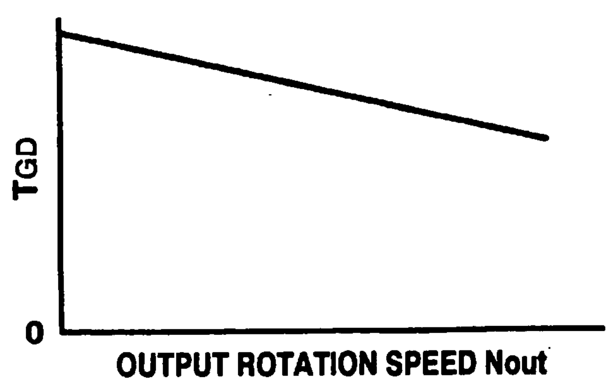


FIG. 13

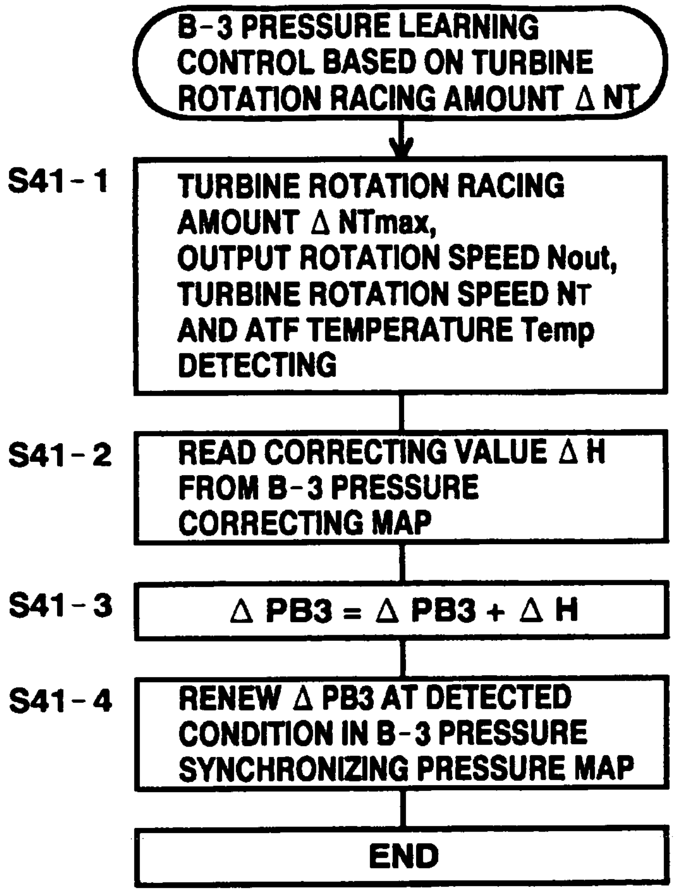

B-3 PRESSURE LEARNING CONTROL BASED ON TURBINE ROTATION RACING AMOUNT Δ NT
S41-1
TURBINE ROTATION RACING AMOUNT Δ NTmax, OUTPUT ROTATION SPEED Nout, TURBINE ROTATION SPEED NT AND ATF TEMPERATURE Temp DETECTING
S41-2
READ CORRECTING VALUE Δ H FROM B-3 PRESSURE CORRECTING MAP
S41-3
Δ PB3 = Δ PB3 + Δ H
S41-4
RENEW Δ PB3 AT DETECTED CONDITION IN B-3 PRESSURE SYNCHRONIZING PRESSURE MAP
END

FIG. 14

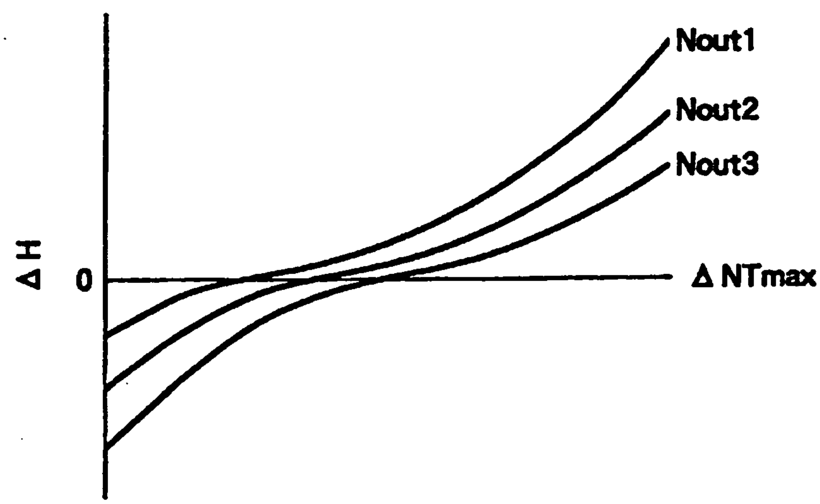

Nout1
Nout2
Nout3
Δ H
0
Δ NTmax

FIG. 15

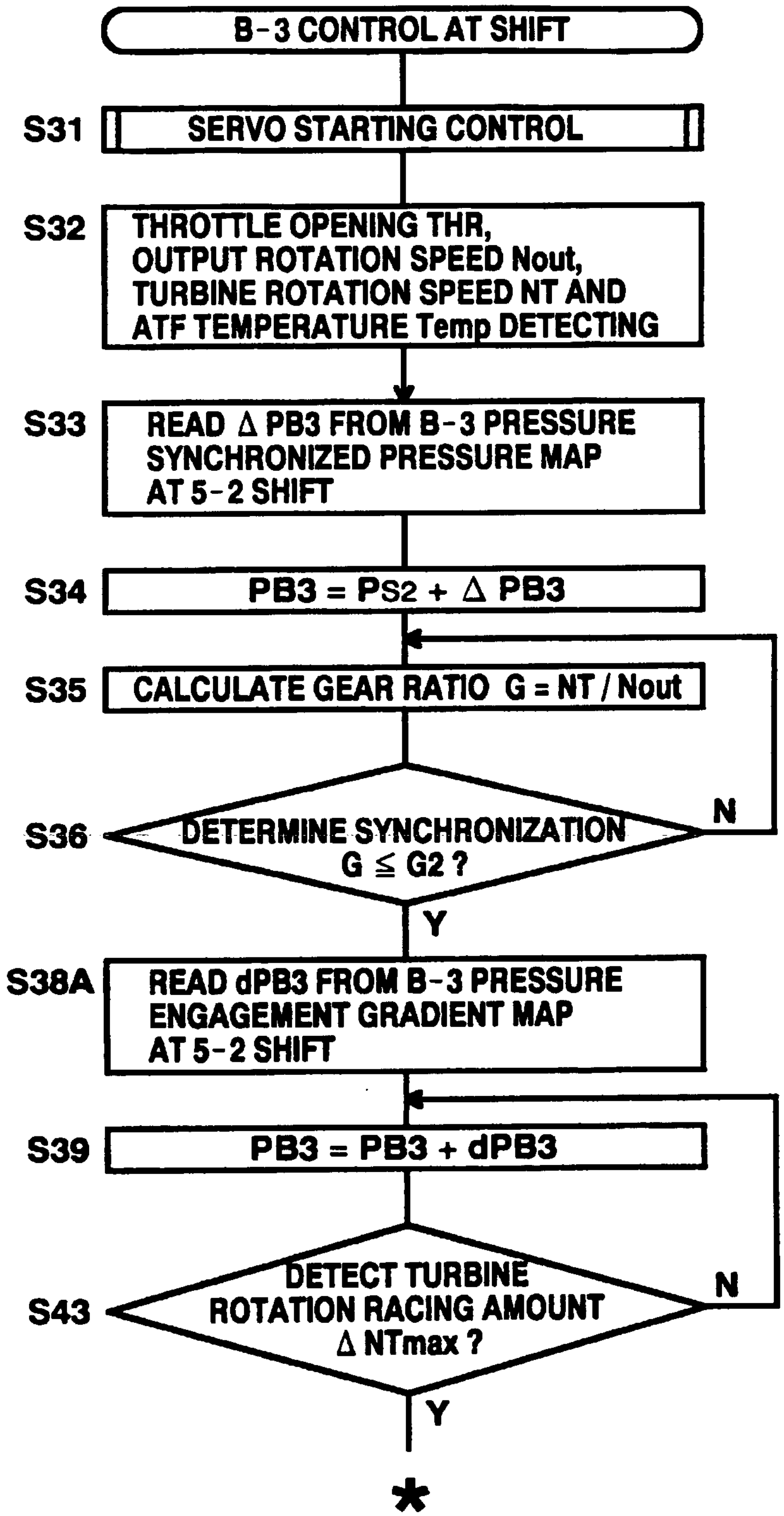

B-3 CONTROL AT SHIFT
S31
SERVO STARTING CONTROL
S32
THROTTLE OPENING THR, OUTPUT ROTATION SPEED Nout, TURBINE ROTATION SPEED NT AND ATF TEMPERATURE Temp DETECTING
S33
READ Δ PB3 FROM B-3 PRESSURE SYNCHRONIZED PRESSURE MAP AT 5-2 SHIFT
S34
PB3 = PS2 + Δ PB3
S35
CALCULATE GEAR RATIO G = NT / Nout
S36
DETERMINE SYNCHRONIZATION G ≦ G2 ?
N
Y
S38A
READ dPB3 FROM B-3 PRESSURE ENGAGEMENT GRADIENT MAP AT 5-2 SHIFT
S39
PB3 = PB3 + dPB3
S43
DETECT TURBINE ROTATION RACING AMOUNT Δ NTmax ?
N
Y
*

FIG. 16

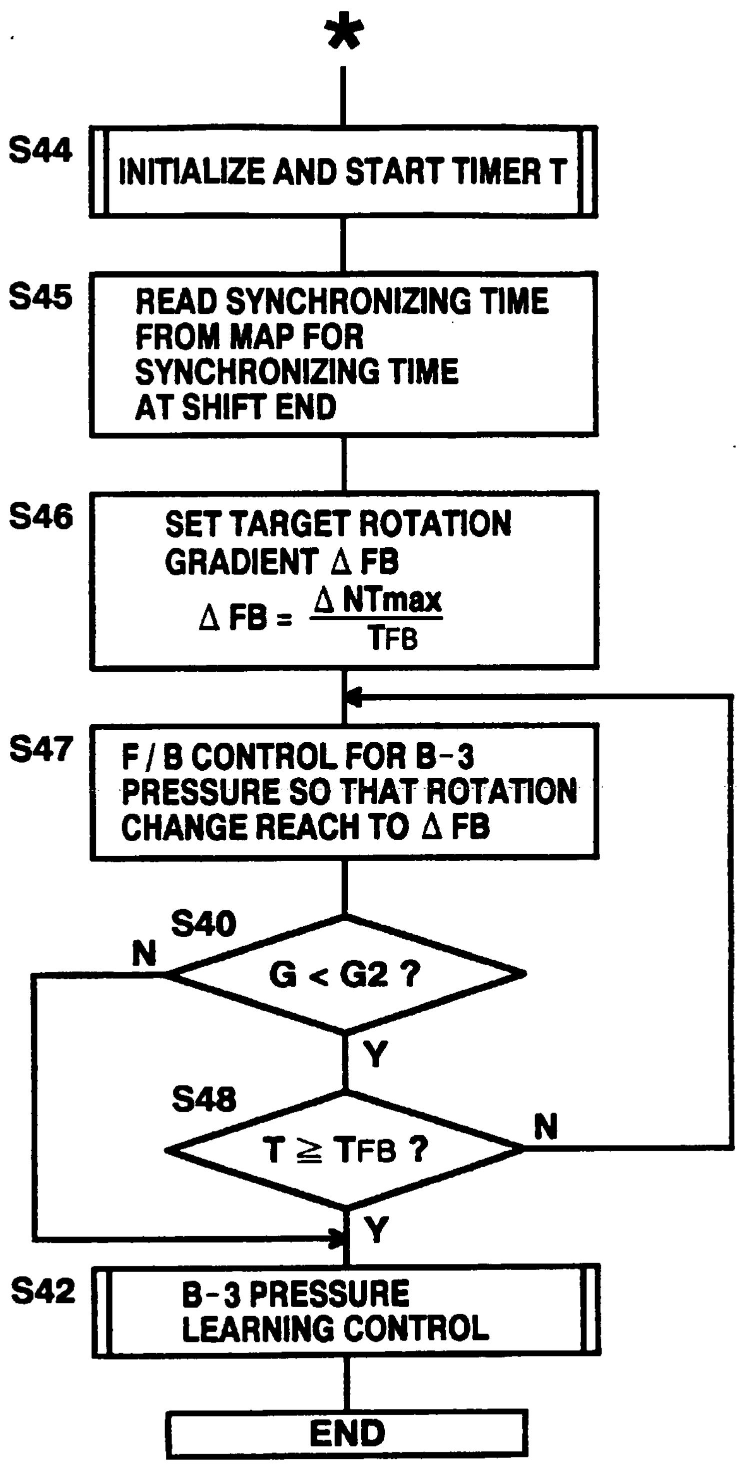

*
S44
INITIALIZE AND START TIMER T
S45
READ SYNCHRONIZING TIME FROM MAP FOR SYNCHRONIZING TIME AT SHIFT END
S46
SET TARGET ROTATION GRADIENT Δ FB
Δ FB = Δ NTmax / TFB
S47
F / B CONTROL FOR B-3 PRESSURE SO THAT ROTATION CHANGE REACH TO Δ FB
S40
G < G2 ?
N
Y
S48
T ≧ TFB ?
N
Y
S42
B-3 PRESSURE LEARNING CONTROL
END

FIG. 17

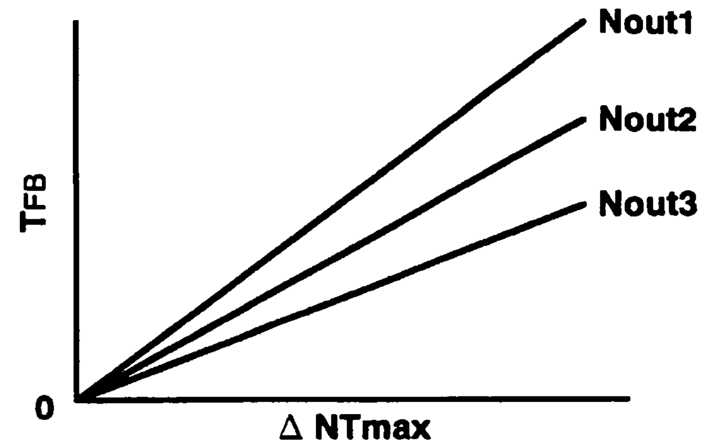
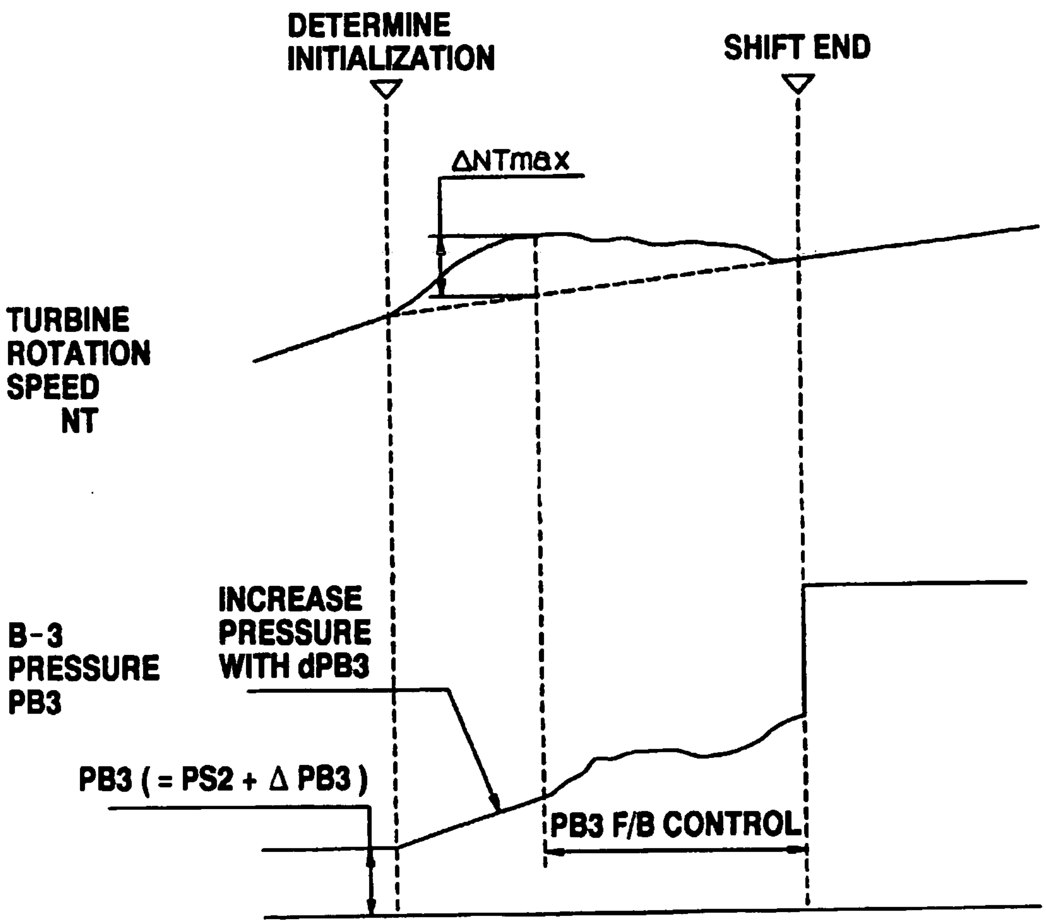

Nout1
Nout2
Nout3
TFB
0
Δ NTmax

FIG. 18


DETERMINE
INITIALIZATION
SHIFT END
ΔNTmax
TURBINE
ROTATION
SPEED
NT
INCREASE
PRESSURE
WITH dPB3
B-3
PRESSURE
PB3
PB3 (= PS2 + Δ PB3)
PB3 F/B CONTROL

CONTROL SYSTEM FOR AUTOMATIC TRANSMISSION

BACKGROUND OF THE INVENTION

1. Field of the Invention

The invention relates to a control system for an automatic transmission. More specifically, it relates to a control system for reducing a shift shock during a down shift.

2. Description of Related Art

An automatic transmission performs gear shift by changing power transmission lines through planetary gear units in a gear train. Frictional engagement elements connected to a particular rotational element of the planetary gear units are properly engaged/disengaged by hydraulic servos so that the power transmission lines are changed. An electronic control system controls a hydraulic control system. The hydraulic control system controls the hydraulic servos.

In this automatic transmission, when the downshift from a high speed gear stage to a low speed gear stage is performed, a frictional engagement element engaged in the high speed gear stage is disengaged and a frictional engagement element engaged in the low speed gear stage is engaged. At that time, when the timing between the disengagement and the engagement is improper, an engine racing or a tie-up shock occur. Japanese patent application laying-open No. 63-266258 describes a feedback control to prevent engine racing and tie-up shock. In this control, when the aforementioned down shift is performed, an input rotation speed of the automatic transmission is changed smoothly at around the synchronizing rotation for the low speed gear stage, then it takes the rotation change of the rotational member into a target rotation change by the feedback control to prevent engine racing and tie-up shock.

In an automatic transmission, the frictional engagement elements are controlled through the hydraulic control system. Therefore, even when the electronic control system controls electrically and minutely, it is difficult to perform proper follow-up control because of a limitation of hydraulic responsiveness and a dispersion of mechanical character of the frictional engagement elements. Especially, in the aforementioned art, the rotation of the rotational member is controlled to achieve the target rotation change set based on the rotation in the shift transition. Therefore, a complicated control is needed.

SUMMARY OF THE INVENTION

In view of the above problems associated with the related art, an object of the invention is to provide a control system for an automatic transmission which prevents engine racing and tie-up shock during a downshift, and in which an effect of hydraulic responsiveness is reduced with a simple structure.

Another object of the invention is to prevent a change of shift character caused by a dispersion of character of each transmission.

Another object of the invention is to achieve a proper shift character corresponding to an operation state of an automatic transmission with a simple control.

Another object of the invention is to prevent a shock caused by steeply engaging the frictional engagement element and an extension of a shift time during a downshift irrespective of a transmission torque of an automatic transmission.

Another object of the invention is to reduce a shift shock based on a difference of vehicle speeds during a downshift.

In order to achieve the aforementioned objects, a control system for an automatic transmission of the invention comprises a frictional engagement element which is engaged to establish a high speed gear stage, a rotational element which is engaged to establish a low speed gear stage and of which the rotational direction at establishing the high speed gear stage is opposite to an operational direction of a reaction torque to the rotational element at establishing the low speed gear stage, a brake that stops the rotational element from rotating, and a control unit which controls the hydraulic pressures for the frictional engagement element and the brake.

The brake is structured from a band brake which has a difference of the engagement force based on a self-energizing operation and a de-energizing operation. The band brake is set so that the direction of the self-energizing operation is the same with an operational direction of a reaction torque from the rotational element at establishing the low speed gear stage. Further, the band brake is set in order to have a region in which a hydraulic pressure, which is needed to engage the rotational element when the low speed gear stage is established, is lower than a hydraulic pressure, which is needed to engage the rotational element when the high speed gear stage is established.

The control unit comprises a disengaging device which disengages the frictional engagement element at the downshift from the high speed gear stage to the low speed gear stage, a synchronization determining device which determines that the input rotation of the automatic transmission is synchronized with the rotation at the low speed gear stage, a constant pressure maintaining device which maintains the hydraulic pressure applied to the hydraulic servo of the band brake with a waiting pressure, which is lower for a predetermined amount than a pressure stopping the rotation of the rotational element in the direction of the self-energizing operation and with which the input rotation is raced for a basic racing amount after a synchronizing point, until the synchronization is determined by the synchronization determining device, and a pressure increasing device which increases the hydraulic pressure applied to the hydraulic servo from the waiting pressure in order to stop the rotational element from rotating after determining the synchronization.

The control unit stores a premeditated waiting pressure value to provide the basic racing amount, and comprises a maximum racing amount detecting device which detects a maximum racing amount of the input rotation speed based on the input rotation speed of the automatic transmission. The control unit comprises a learning device which renews the stored waiting pressure value in order to achieve the basic racing amount by comparing the basic racing amount with the maximum racing amount after the end of the downshift.

The multiple waiting pressure values are stored in the control unit corresponding to the throttle opening of the engine and the output rotation speed as parameters, and the waiting pressure value is selected corresponding to the throttle opening and the output rotation speed at the shift start.

The pressure increasing device increases the hydraulic pressure with a predetermined rate, which increases when the throttle opening of the engine increases, after determining the synchronization.

The pressure increasing device increases the hydraulic pressure at a predetermined rate and controls the hydraulic pressure in order that the input rotation speed changes at a

target rate, which reduces when the output rotation speed of the automatic transmission reduces.

According to the invention, the rotation of the rotational element reduces to synchronize with the rotation of the rotational element at the low speed gear stage. That is, the rotation of the rotational element reduces to stop. In this case, the de-energizing operation occurs at the band brake. Therefore, the rotational element is not stopped from rotating by the band brake, because the engagement force occurred by the application of the aforementioned hydraulic pressure is small. After that, when the rotational element is stopped from rotating and then the reverse rotation of the rotational element is started, the self-energizing operation occurs. Therefore, the engagement force of the band brake steeply increases to stop the rotational element from rotating.

In this case, the hydraulic pressure applied to the hydraulic servo of the band brake is the waiting pressure, which is lower for the predetermined amount than the hydraulic pressure to maintain the stop of the rotation of the rotational element. Therefore, the rotational element is not steeply stopped, that is, the rotation of the rotational element changes gradually.

rapid -happens 1st

2nd control stage

Then, after determining the synchronization, the hydraulic pressure increases to stop the rotational element from rotating, and then the shift is ended. Therefore, the responsibility of the hydraulic pressure is not made to be a problem, because the characteristic of the band brake changes and the engagement force increases corresponding to the rotational state of the rotational element by setting the hydraulic pressure as above. That is, a control to change the hydraulic pressure is not needed. Further, the rotation of the rotational element changes gradually at around the synchronizing rotation without a feedback control performed based on the rotation speed before the synchronization. Therefore, large engine racing and large tie-up shock are prevented with a simple control.

Even when each automatic transmission has dispersion or the band brake and the operational fluid of the automatic transmission deteriorate with age, the racing amount is always controlled properly to reduce shift shock because the waiting pressure value, with which the basic racing amount of the rotational element is determined, is renewed based on the actual maximum racing amount.

It is preferable that the waiting pressure is set based on the input torque and the inertia torque of the transmission at the shift end to reduce the shift shock. In this case, a complicated calculation will be needed to exactly calculate these torque values. In the invention, the multiple waiting pressure values are stored based on the throttle opening of the engine and the output rotation speed of the automatic transmission. Therefore, the waiting pressure corresponds to the input torque based on the throttle opening and the output rotation speed, and to the inertia torque based on the output rotation speed. As a result, the waiting pressure is made to be proper with a simple method.

The steep engagement at the low throttle opening and the extension of the shift time at the high throttle opening are prevented by increasing the hydraulic pressure at the predetermined rate, which increases when the throttle opening increases, after determining the synchronization.

A rate of the turbine rotation maximum racing amount to the whole rotation change amount in the shift in the low vehicle speed is higher than the case of the high vehicle speed. Therefore, a steep rotation change to the whole rotation change in the shift is prevented. Then, a shift shock is more certainly prevented by setting the gradient to reduce when the output rotation speed reduces.

BRIEF DESCRIPTION OF THE DRAWINGS

The invention will be described in conjunction with the following drawings in which like features are designated with like reference characters, and wherein:

FIG. 1 is a circuit diagram showing a first embodiment of the control system for an automatic transmission of the invention;

FIG. 2 is a system structuring diagram showing the entire structure of an automatic transmission provided with the shift mechanism of the invention, which is shown schematically;

FIG. 3 is an operation diagram of the automatic transmission;

FIG. 4 is a chart showing a needed engagement force of a brake of the automatic transmission to establish a low speed gear stage;

FIG. 5 is a time chart for a shift control of the first embodiment;

FIG. 6 is a main flowchart for the shift control by the control system;

FIG. 7 is a flowchart for a subroutine of a B-2 control during the shift by the control system;

FIG. 8 is a flowchart for a subroutine of a B-3 control during the shift by the control system;

FIG. 9 is a flowchart for a subroutine of a servo starting control in the control system;

FIG. 10 is a chart of a synchronizing pressure map for a B-3 pressure in the control system;

FIG. 11 is a chart of an engagement gradient map for a B-3 pressure in the control system;

FIG. 12 is a chart of a synchronized time guard timer map in the control system;

FIG. 13 is a flowchart for a subroutine of a B-3 pressure learning control in the control system;

FIG. 14 is a chart of a B-3 pressure correction map in the control system;

FIG. 15 is a flowchart for the first half of a subroutine of a shift time B-3 control in a shift control of the second embodiment;

FIG. 16 is a flowchart for the last half of a subroutine of a shift time B-3 control;

FIG. 17 is a shift end time synchronized time map in the control system of the second embodiment; and

FIG. 18 is a time chart of the last half of the shift in the control system of the second embodiment.

DETAILED DESCRIPTION OF PREFERRED EMBODIMENTS

The invention will become more apparent from the detailed description of preferred embodiments with reference to the accompanying drawings.

FIG. 2 shows an automatic transmission having a control system of the invention. The structures of the automatic transmission are described below.

The automatic transmission comprises a torque converter 12 having a lock-up clutch L/C and interlocking to a vehicular engine E/G, a shift mechanism having three planetary gear sets M1, M2, M3 that shift the output from the torque converter 12 to five forward speeds and one back-

ward speed, and differential unit **21** interlocking through a counter gear **20**, which is also a speed reducing mechanism, to the shift mechanism, and transmitting the reduced output to left and right wheels of the vehicle.

In the shift mechanism of the automatic transmission, pinion gears **p1** and **p2**, which have different diameters each other, of the gear sets **M1** and **M2**, are directly connected each other, a ring gear **r1** of the gear set **M1** is connected to a carrier **c3** of the gear set **M3**, a ring gear **r3** of the gear set **M3** is connected to a carrier **c1** of the gear set **M1**, and a sun gear **s1** and a carrier **c1** of the gear set **M1** are input members and connected through clutches **C-1** and **C-2**, respectively, to an input shaft **14**.

The input shaft **14** is interlocked to a turbine shaft **13** of the torque converter **12**. The ring gear **r1** and carrier **c3** interlocking each other are interlocked to an output gear **19**. The sun gear **s1** of the gear set **M1** is able to be engaged through a brake **B-1** with a transmission case **10**. The sun gear **s2** of the gear set **M2** is able to be engaged through a brake **B-2** with the transmission case **10**. The sun gear **s3** of the gear set **M3** is able to be engaged through a brake **B-3** with the transmission case **10**. The ring gear **r3** interlocking to the carrier **c1** is able to be engaged through a brake **B-R** with the transmission case **10**. The output gear **19**, as an output member, is interlocked through the counter gear **20** to the differential unit **21**. The brakes, except for the brake **B-R**, are band brakes. The brake **B-R** is a multiple disc clutch. The hydraulic servos for the brakes are not shown in the figure.

The automatic transmission thus structured establishes each gear stage by selectively engaging and disengaging the clutches and the brakes, as shown in FIG. **3**. In FIG. **3**, circles constitute engagement. The clutches and the brakes are engaged and disengaged by applying/draining hydraulic pressures to/from the hydraulic servos according to the clutches and the brakes based on a control by an electronic control system **6** and a hydraulic control system **5**.

The first gear stage is established by engaging the clutch **C-1** and the brake **B-3**. In this case, the rotation from the input shaft **14** is transmitted through the clutch **C-1** to the sun gear **si**, then outputted as the rotation of the carrier **c3**, which is the most reduced rotation by engaging the brake **B-3** that stops the sun gear **s3** from rotating, to the output gear **19**. The second gear stage is established by engaging the clutch **C-2** and the brake **B-3**. In this case, the rotation from the input shaft **14** is transmitted through the clutch **C-2** and the carrier **c1** to the ring gear **r3**, then outputted as the differential rotation of the carrier **c3**, which serves as a reaction element reacting to the sun gear **s3** stopped from rotating by engaging the brake **B-3**, to the output gear **19**. The third gear stage is established by the directly connecting state of the planetary gear set **M1** achieved by engaging clutches **C-1** and **C-2**. In this case, the rotation from the input shaft **14** is outputted as the rotation of the carrier **c3** to the output gear **19**.

The fourth gear stage and above of the transmission operate as an overdrive. The fourth gear stage is established by engaging the clutch **C-2** and engaging the brake **B-1** stopping the sun gear **s1** from rotating. In this case, the rotation from the input shaft **14** is transmitted to the output gear **19** through the carrier **c3** as the rotation of the ring gear **r1**, which is accelerated by the rotation of the pinion gears **p1** with respect to the rotation of the carrier **c1**. The fifth gear stage is established by engaging the clutch **C-2** and the brake **B-2**. In this case, the rotation from the input shaft **14** is transmitted to the output gear **19** through the carrier **c3** as the rotation of the ring gear **r1**, which is additionally accelerated by the rotation of the small-diameter pinion gear **p2**, reacting the sun gear **s2** having a larger diameter than the sun gear **s1**, with respect to the rotation of the carrier **c1**.

The reverse gear stage is established by engaging the clutch **C-1** and the brake **B-R**. In this case, the rotation from the input shaft **14** is transmitted through the clutch **C-1** to the sun gear **s1**, the rotation of the carrier **c1** is stopped by engaging the ring gear **r3** with the case **10** due to the engagement of the brake **B-R**, and the reverse rotation of the ring gear **r1**, which is decelerated by the rotation of the pinion gear **p1**, is outputted through the carrier **c3** to the output gear **19**.

In the automatic transmission thus structured, the brake **B-2** is the frictional engagement element which is engaged to establish the high speed gear stage, the sun gear **s3** is the rotational element which is stopped from rotating to establish the low speed gear stage, and the brake **B-3** is the brake to stop the rotation of the sun gear **s3**. The control system controlling the hydraulic pressure to the hydraulic servos of the brakes **B-2** and **B-3** is structured as a circuit within the hydraulic control system **5** and a program stored in the electronic control system **6** controlling the circuit with electric signals.

As shown in FIG. **1** in detail, the brake **B-3** comprises a drum **31**, a band **32**, and a hydraulic servo **4**. The drum **31** is interlocked to the sun gear **s3**. The band **32** comprises brackets **33** and **34** at each end on the outer periphery of the band **32**. The anchor side bracket **33** is attached to an anchor pin **35** which is fixed to the case **10**. An apply side bracket **34**, on the pressure application side, is attached to the end of a piston rod **42** of the hydraulic servo **4**. The elasticity of the band **32** in a direction for opening, i.e., toward the hydraulic servo **4**, biases the bracket **34** to the piston rod **42**. Because of the characteristics of the mechanism of the brake **B-3**, at the engagement of the brake **B-3**, when the drum **31** receives a counterclockwise torque as shown in the FIG. **1**, the frictional force that occurs by engaging the band **32** with the drum **31** acts on the band **32** in a direction to further tighten the band **32**. Therefore, a selfenergizing (referred to as self-energy hereafter), which increases the brake engaging forces, occurs. When the drum **31** receives a clockwise torque, the frictional force that occurs by engaging the band **32** with the drum **31** acts on the band **32** in a direction to release the band **32**. Therefore, a de-energizing (referred to as de-energy hereafter), which reduces the brake engaging forces, occurs. As a result, the engagement forces of the brake **B-3** are different based upon the direction of the reacting torque applied on the sun gear **s3**.

The hydraulic servo **4** of the brake **B-3** comprises a servo cylinder **40** having cylinder bores S_L, S_S which have different diameters, a large diameter piston **44** slidably inserted in the large bore S_L, a small diameter piston **43** slidably inserted in the small bore S_S, a rod **42** inserted through both of the pistons **43** and **44** and always seated against or in contact with the small diameter piston **43**, a separator spring **45** and a return spring **46** which are formed from compressed coil springs having different diameters, and a lid **41** covering an opening in the end of the large bore S_L. The rod **42** fixed against the small diameter piston **43** slidably protrudes through an end wall on the small bore S_S side of the servo cylinder **40** and is attached to the bracket **34** of the band **32**. The large diameter piston **44** is slidably retained by the rod **42**. The separator spring **45** having a smaller diameter than the return spring **46** is arranged with a predetermined load setting between the small diameter piston **43** and the large diameter piston **44**. The return spring **46** having a larger diameter than the separator spring **45** is arranged with a

predetermined load setting between the end wall of the servo cylinder **40** and the large diameter piston **44**.

The hydraulic control system **5** controlling the hydraulic servo **4** comprises a hydraulic pressure source **50** having a pump as a main body for the line pressure P_L, a B-2 control valve **51** connected through a line pressure hydraulic path p to the hydraulic pressure source **50**, modulating the line pressure P_L and outputting the modulated pressure to the hydraulic servo of the brake B-2, a B-3 control valve **52** connected to the line pressure hydraulic path p, modulating the line pressure P_L and outputting the modulated pressure to the hydraulic servo **4**, a solenoid modulator valve **53** connecting to the line pressure hydraulic path p, reducing the line pressure P_L and outputting the reduced pressure to a modulator pressure hydraulic path m, a linear solenoid valve **54** outputting a solenoid signal pressure, which is based on the modulator pressure Pm reduced at the solenoid modulator valve **53**, through a hydraulic path q to the B-2 control valve **51**, and a linear solenoid valve **55** outputting a solenoid signal pressure, which is based on the modulator pressure Pm reduced at the solenoid modulator valve **53**, through a hydraulic path r to the B-3 control valve **52**.

The electronic control system **6** controlling the linear solenoid valves **54** and **55** are connected to the solenoids of the both valves. Further, as shown in FIG. **2**, the electronic control system **6** is connected to a throttle opening sensor **71**, a turbine rotation speed sensor **72**, output rotation speed sensor **73**, and an automatic transmission fluid (ATF) temperature sensor **74**. The throttle opening sensor **71** is used to determine the shift and to select a map discussed below. The turbine rotation speed sensor **72** is used to detect the transmission input rotation speed to determine the shift start and the synchronization. The output rotation speed sensor **73** is used to detect the vehicle speed for selecting a map data. The ATF temperature sensor **74** is used to select the map.

The band brake B-3 is set to have a region in which a pressure, which is needed to engage the brake drum **31** when the brake drum **31** interlocked to the sun gear s3 rotates in the direction of the reacting force operated to the brake drum **31** (negative direction) at the second gear stage, is lower than a pressure, which is needed to engage the brake drum **31** when the brake drum **31** rotates in the direction of the rotation at the fifth gear stage (positive direction). That is, the de-energy occurs in the band brake B-3 when the brake drum **31** rotates in the positive direction, and the self-energy occurs in the band brake B-3 when the brake drum **31** rotates in the negative direction or when the brake drum **31** is in a stopping state and the torque having the negative direction is operated.

In detail, when the fifth gear stage is established, the rotation, which having the positive direction, inputted through the clutch C-2 to the carrier c1 is accelerated by stopping the sun gear s2 from rotating and outputted from the ring gear rl. Then, the rotation from the ring gear r1 is outputted through the carrier c3 to the output gear **19**. In this case, the sun gear s3 rotates in the positive direction because the ring gear r3 rotates with the input rotation the same as the carrier c1 and the carrier c3 rotates with the accelerated rotation, which is higher than the input rotation, the same as the ring gear r1. Therefore, the brake drum **31** rotates in the positive direction. When the second gear stage is established, the rotation inputted through the clutch C-2 and the carrier c1 to the ring gear r3 is decelerated by stopping sun gear s3 from rotating and outputted from the carrier c3 to the output gear **19**. In this case, the sun gear s3 receives the reaction force having the negative direction. Therefore, the brake drum **31** receives this reaction force having the negative direction.

The band brake B-3 has the aforementioned region when the output shaft is driven by the drive torque from the engine, that is, when the vehicle is in a power on state in which an accelerator pedal is depressed, and when the accelerator pedal is released, the vehicle is driven with a rather low speed and the engine rotates with less rotation speed than the idle rotation. The band brake B-3 does not have the aforementioned region when the engine is driven by the drive torque from the output shaft, that is, when the vehicle is in a coast state, because the direction of the reaction force reacting to the brake drum **31** changes to the positive direction. Therefore, this invention is applied in the case in which the output shaft is driven by the drive torque from the engine.

This setting is described with reference to FIG. 4. FIG. **4** is a chart that shows the hydraulic pressure needed to engage the band brake at a predetermined output rotation speed. As shown in FIG. **4**, the hydraulic pressure, which is needed to completely stop the rotational element at the rotation in the self-energy direction as shown with a solid line, is lower than the pressure, which is needed at the rotation in the de-energy direction as shown with a solid line. As a result, the reverse rotation of the rotational element is able to be stopped by the hydraulic pressure needed for stopping the rotation in the self-energy direction as a one-way clutch. However, in case that the hydraulic pressure is set as shown with a dotted line wherein a certain engine racing occurs at the end of the shift, the steep brake engagement at the reverse rotation of the rotational element is prevented so that the smooth shift is performed. In effect, it is very complicated to calculate the input torque and the inertia torque at the engagement of the brake, that is, at shift end. Therefore, in this embodiment, an easy control is performed by storing a data, which sets a relation between an output rotation speed Nout and a waiting pressure value APB3 with a throttle opening THR as a parameter, as shown in a B-3 pressure synchronization pressure map for the 5–2 shift as shown in FIG. **10** discussed below.

An apparatus that performs the control includes a program stored in the electronic control system **6**. The control performed by the program will be conceptually described with reference to a time chart. FIG. **5** shows the time chart for a 5–2 shift control which is an example of a down shift. In this case, at first, a B-2 pressure PB2 applied to the hydraulic servo of the brake B-2 is an engagement pressure according to the input torque, and a B-3 pressure PB3 applied to the hydraulic servo of the brake B-3 is zero which means a releasing state. A turbine rotation speed NT is a low rotation speed synchronized with the rotation of the fifth gear stage. The throttle opening THR is zero which means a throttle off state. An output torque Tout is a negative value which means an engine coasting state. A B-3 drum rotation speed NB3 is in an idle running state, and the direction of the rotation is the de-energy direction.

When the throttle opening THR is increased by, for example, a kick-down, the control is started based on the 5–2 shift determination by the electronic control system **6**. The B-2 pressure PB2 is increased to a high value by an increase of a throttle pressure due to the increase of the throttle opening so that the engagement of the brake B-2 is maintained according to the increase of the torque. Then, the high value is maintained for a predetermined time by a timer control so that the engine racing due to the under lap of the both brakes B-2 and B-3 is prevented. In this case, the output torque Tout changes to a positive value because the engine state changes to the engine drive state. The B-3 pressure PB3 increases to a first fill pressure P_{S1} to fill the clearance of the

hydraulic servo piston and maintained at the value for a predetermined time t_{SA}.

When the predetermined time t_{SA} to fill the clearance has elapsed, the B-3 pressure PB3 is reduced to a stroke pressure P_{S2} at a predetermined rate. Then, the B-3 pressure PB3 is maintained at the stroke pressure P_{S2} until a predetermined time t_{SE} has elapsed. The B-2 pressure PB2 is drained at a predetermined rate after elapsing the predetermined time t_{SA}. According to the start of the actual shift due to the drain of the B-2 pressure PB2, the turbine rotation speed NT as the input rotation speed increases to the neutral rotation speed, and the B-3 drum rotation speed NB3 reduces because of the reverse of the reaction force.

When the predetermined time t_{SE} has elapsed, the B-3 pressure PB3 increases for an amount of the waiting pressure value ΔPB3, and the B-3 pressure PB3 is maintained at the value. In this state, the shift proceeds. Then, when the synchronization is determined, the B-3 pressure PB3 increases at a predetermined rate dPB3. At that time, the B-3 drum rotation speed reduces to zero and starts the reverse rotation immediately. However, the brake B-3 is not engaged immediately because the engagement force of the band is lacked due to the low hydraulic pressure setting of the hydraulic control. Therefore, the turbine rotation speed NT is raced for a certain amount comparing with the synchronizing rotation of the second gear stage as shown with a dotted line, after that, the turbine rotation speed NT reduces gradually, then, achieves to the synchronizing rotation at the shift end.

The difference between the turbine rotation speed NT at the synchronizing rotation and the turbine rotation speed NT at starting to reduce is defined to be a maximum turbine racing amount (ΔNTmax). In this term, the B-3 drum rotates with a certain speed in the self-energy direction, then, the B-3 drum rotation speed gradually reduces to zero, which means the stopping state, at the shift end by the increase of the engagement force of the band due to the sweep-up of the B-3 pressure PB3. At that time, the B-3 pressure PB3 immediately increases to the line pressure to ensure the maintenance of the engagement state, then, the shift is ended.

FIG. **6** is a main flowchart of the hydraulic control process performed in the electronic control system **6** to perform the aforementioned control at the 5–2 downshift. In this flowchart, the control is divided to a control at shift and a usual control by the determination of the 5–2 shift. At step **S1**, it is determined whether the 5–2 shift is determined. When the 5–2 shift is not determined, the usual control is performed at step **S4**. When the 5–2 shift is determined, the subroutine for the B-2 control at shift is performed at step **S2**, and the subroutine for the B-3 control at shift is performed at step **S3**.

FIG. **7** shows details of the B-2 control at shift of the brake B-2 which is disengaged. At step **S21**, the throttle opening THR, the output rotation speed Nout, the turbine rotation speed NT and the ATF temperature Temp are detected by the outputs from the aforementioned sensors respectively. At step **22**, a timer T_2 that maintains the B-2 pressure PB2 for the predetermined time is initialized. At step **S23**, the B-2 pressure PB2 is maintained. At step **S24**, the elapse of the timer T_2 is waited ($T_2 \geqq t$, t: constant time). After elapsing the timer T_2, at step **S25**, the B-2 pressure PB2 reduces at the predetermined rate, then, the process is ended.

Regarding the brake B-3 which is engaged, the B-3 control at shift shown in FIG. **8** is performed. In this control, at step **S31**, a servo starting control is performed. This starting control is shown in FIG. **9**. At step **S31-1**, a timer for the timer control is started (t=0). At step **S31-2**, the process, in which the application pressure PB3 is set to be the first fill pressure P_{S1} to fill the invalid stroke of the piston, is performed. In detail, the output to the linear solenoid valve **55** shown in FIG. **1** is set to be a duty ratio with which the output pressure from the B-3 control valve **52** achieves to the first fill pressure P_{S1}. At step **S31-3**, it is determined whether the timer t has elapsed the predetermined time t_{SA}. When the timer t has elapsed the predetermined time t_{SA}, at step **S31-4**, the application pressure PB3 reduces at the predetermined rate $(P_{S1}-P_{S2})/t_{SB}$. This process continues until the application pressure PB3 reduces to lower than the stroke pressure P_{S2} at step **S31-5**. When the application pressure PB3 reduces to lower than the stroke pressure P_{S2}, at step **S31-6**, the application pressure PB3 is maintained at the stroke pressure P_{S2}. This state continues until the timer t has elapsed the predetermined time t_{SE} ($t>t_{SE}$) at step **S31-7**. Then, the servo starting control is ended.

Returning to FIG. **8**, after the servo starting control, at step **S32**, the throttle opening THR, the output rotation speed Nout, the turbine rotation speed NT, and the ATF temperature Temp are detected. At step **S33**, the waiting pressure value ΔPB3 is read from the synchronizing pressure map for the B-3 pressure at the 5–2 shift. FIG. **10** shows the synchronizing pressure map for the B-3 pressure at the 5–2 shift. In this map, the waiting pressure value ΔPB3 is defined with the relation to the output rotation speed Nout and with the throttle opening THR as the parameter.

As shown in FIG. **10**, when the throttle opening THR reflecting the input torque increases, the waiting pressure value ΔPB3 increases. In this map, the input torque is presumed based on the throttle opening THR and the vehicle speed, that is, the output rotation speed Nout, and the inertia torque is presumed based on the output rotation speed Nout. Therefore, the map corresponds to the change of the input torque and the inertia torque. It notes that it is better that the multiple maps corresponding to the ATF temperature Temp are prepared and the multiple maps are set so that the waiting pressure value ΔPB3 increases when the ATF temperature Temp reduces in case the μ-characteristic reduces when the ATF temperature Temp reduces. Further, it is better that the maps are set properly corresponding to the materials of the frictional engagement element because the change of the μ-characteristics corresponding to the ATF temperature Temp are different depending upon the material.

Returning to FIG. **8**, at step **S34**, the application pressure PB3 is set based on the waiting pressure value ΔPB3 and the stroke pressure P_{S2} and outputted as the waiting pressure. At step **S35**, the gear ratio G is calculated by dividing the turbine rotation speed NT by the output rotation speed Nout. At step **S36**, the second gear stage synchronization is determined based on whether the calculated gear ratio G reduces to the gear ratio G2 of the second gear stage ($G \leqq G2$). When the synchronization is determined, at step **S37**, a timer T_1 is initialized and started. At step **S38**, the engagement rate dPB is read from an engagement rate map for the B-3 pressure at the 5–2 shift shown in FIG. **11**, and a synchronizing time guard timer T_{GD} is read from a map shown in FIG. **12**.

FIG. **11** shows the engagement rate map for the B-3 pressure at the 5–2 shift. According to this map, the engagement rate dPB is made to be large when the throttle opening, that is, the input torque is large so that the extension of the shift time is prevented, and the engagement rate dPB is made to be small when the throttle opening is low so that the shift shock is prevented.

FIG. 12 shows the map of the synchronizing time guard timer to prevent the baking of the frictional element because of the long shift time. At step S39, the engagement rate thus obtained is added to the current application pressure PB3 and the renewed application pressure PB3 is outputted. At step S40, the synchronization is determined (G<G2). When the synchronization is determined, at step S42, a B-3 pressure learning control is performed. In case the synchronization is not determined until the timer T_1 has elapsed the guard timer T_{GD}, after the determination of that the timer T_1 has elapsed the guard timer T_{GD} at step S41, the B-3 pressure learning control is performed at step S42.

FIG. 13 shows the detail of the B-3 pressure learning control. At step S41-1, the turbine rotation maximum racing amount ΔNTmax, the output rotation speed Nout, the turbine rotation speed N_T, and the ATF temperature Temp are detected. The turbine rotation maximum racing amount ΔNTmax is the maximum value of the difference between the actual input rotation speed and the synchronizing input rotation speed, which is calculated based on the output rotation speed Nout and the gear ratio, at the low speed gear stage. At step S41-2, a correcting value ΔH is read with reference to a B-3 pressure correcting map shown in FIG. 14. At step S41-3, the waiting pressure value ΔPB3 is renewed by adding the correcting value ΔH and outputted. At step S41-4, the waiting pressure value ΔPB3 of the detected condition in the B-3 pressure correcting map is renewed as shown with a dotted line in FIG. 10.

FIG. 14 shows the B-3 pressure correcting map. In this map, a point crossing with a line of zero of the correcting value ΔH is the basis of the turbine rotation maximum racing amount ΔNTmax. When the racing amount is larger than the basis, the hydraulic pressure increases to prevent the extension of the shift time. When the racing amount is smaller than the basis, the hydraulic pressure reduces to prevent the shift shock. In case the basis is set to be constant irrespective of the vehicle speed, that is, the output rotation speed Nout, the racing feel increases at the low vehicle speed because whole rotation change rate at the shift in the low vehicle speed is smaller than the rate in the high vehicle speed. Therefore, the basis is needed to set in order that the basis reduces when the vehicle speed reduces to prevent the large racing feel. Therefore, in this map, the correcting value ΔH increases when the vehicle speed reduces in the region larger than the basis.

In the process performed by the electronic control system 6, in the hydraulic circuit shown in FIG. 1, a solenoid pressure, which is modulated from a modulator pressure Pm at the linear solenoid valve 55, is applied to an end of a spool of a B-3 control valve 52. The B-3 control valve 52 modulates the line pressure P_L to the waiting pressure ($PB3=P_{S2}+\Delta PB3$) with the balance between a return spring load against the force of the solenoid pressure and a feedback pressure. Then, the application pressure PB3 maintained at the constant waiting pressure is applied to the large diameter bore S_L of the hydraulic servo 4. The large diameter piston 44 is slid and the rod 42 is pushed by the application of the constant waiting pressure. Then, the rod 42 pushes the bracket 34 at the end. At that time, the band 32 supported by the anchor pin 35 at the end is engaged with the drum 31. However, in a state before the synchronization of the second gear stage in which the reaction force operated to the drum 31 in the engagement state operates in the de-energizing direction, the rotation speed reduces but the drum 31 continues to rotate because the engagement force of the band 32 is lacked. After that, the rotation speed of the drum 31 reduced to zero when the synchronizing point of the second gear stage is achieved. However, at that time, the engagement force of the band 32 is lacked because the application pressure is the waiting pressure $PB3=P_{S2}+\Delta PB3$. Therefore, the drum 31 rotates in the reverse direction. After that, the application pressure PB3 is increased at the predetermined rate dPB3 by the electronic control system 6. Therefore, the application pressure PB3 increases to a hydraulic pressure with which the rotation in the self-energizing direction is stopped. As a result, the rotation of the drum 31 is stopped from rotating.

In this embodiment, when the sun gear s3 is engaged to establish the second gear stage, the hydraulic pressure, which is needed to stop the rotation of the sun gear s3 rotating in the de-energizing direction, is higher than the hydraulic pressure in case of the self-energizing direction. The self-energizing direction is set to be the same with the direction in which the drum 31 rotates by the reaction force to the sun gear s3 occurred at establishing the low speed gear stage. The hydraulic pressure, with which only the rotation in the self-energizing direction is stopped, is applied from the synchronization of the fifth speed gear stage. Therefore, the rotational direction of the drum 31, which is in the de-energizing direction at establishing the high speed gear stage, changes to the self-energizing direction at establishing the low speed gear stage, then, the second gear stage is established by engaging the sun gear s3 which is engaged by the band brake B-3 with the self-energizing operation. In case that the hydraulic pressure, with which the drum 31 is stopped immediately at the synchronizing point, is applied, the shift shock increases because of the stop of the rotation of the sun gear s3 at the synchronizing point. Therefore, in this embodiment, the reverse rotation of the sun gear s3 is stopped gradually after the reverse rotation occurs which occurs after the synchronizing point.

FIG. 15 and FIG. 16 show a control flowchart of the second embodiment of the invention. This embodiment is different from the first embodiment such that a feedback control is performed at the latter of the sweep-up of the B-3 pressure. Therefore, the features in common with the first embodiment are not described and the same step numbers are used. Only the different control features will be described. In this feedback control, after determining the synchronization at step S36 shown in FIG. 15, without initializing and starting the timer T_1, the engagement gradient pressure dPB3 is read from the map at step S38A, and the sweep-up of the B-3 pressure is performed at the rate of the engagement gradient pressure dPB3 at step S39. Then, at step S43, the turbine rotation maximum racing amount ΔNTmax is detected.

As shown in FIG. 16, at step S44, a timer T is initialized and started. At step S45, a synchronizing time T_{FB} is read from a map for a synchronizing time at the shift end.

FIG. 17 shows the map for the synchronizing time at the shift end. As shown in FIG. 17, the shift time increases when the turbine rotation maximum racing amount ΔNTmax increases in order to prevent the shift shock by a steep rotation change. A rate of the turbine rotation maximum racing amount ΔNTmax to the whole rotation change amount in the shift in low vehicle speed is higher than the case of high vehicle speed. Therefore, the shift time is lengthened when the output rotation speed reduces in order to prevent the steep rotation change to the whole rotation in the shift.

Returning to FIG. 16, at step S46, a target rotation change ΔFB is set. The target rotation change ΔFB is calculated by dividing the turbine rotation maximum racing amount

ΔNTmax with the synchronizing time T_{FB}. At step S47, the feedback control for the B-3 pressure PB3 is performed so that the actual rotation change achieves to the target rotation change ΔFB. Step 40 is same step as the first embodiment. At step S48, it is determined whether the timer T has elapsed the synchronizing time T_{FB}. Step 42 is same step as the first embodiment.

FIG. 18 shows a time chart after determining the synchronization of the second embodiment. When the turbine rotation maximum racing amount ΔNTmax is detected, the feedback control starts. Therefore, the B-3 pressure PB3 is controlled based on the feedback value.

The control of the second embodiment is complicated in comparison with the control of the first embodiment. However, the steep rotation change to the whole rotation change in the shift is prevented, then, the shift shock is more certainly prevented by setting the gradient to reduce when the output rotation speed reduces corresponding to that a rate of the turbine rotation maximum racing amount ΔNTmax to the whole rotation change amount in the shift in the low vehicle speed which is higher than the case of the high vehicle speed.

The invention should not be limited to the foregoing embodiments but can be modified in various manners based on its gist, and these modifications should not be excluded from the scope of the invention.

What is claimed is:

1. A control system for an automatic transmission that has a high speed gear stage, a low speed gear stage, and an input rotation, comprising:

- a frictional engagement element engageable to establish the high speed gear stage;
- a rotational element that can be stopped from rotating with a brake to establish the low speed gear stage, a rotational direction of the rotational element when the high speed gear stage has been established being opposite to a direction of torque to the rotational element created when the rotational element is stopped from rotating to establish the low speed gear stage; and
- a control unit that controls hydraulic pressures applied to the frictional engagement element and the brake, wherein:
  - the brake is formed of a band brake which has a difference of engagement force between a self-energizing operation and a de-energizing operation, and includes a hydraulic servo;
  - the band brake is set so that a direction of the self-energizing operation is the same as the direction of the torque to the rotational element created when the rotational element is stopped from rotating to establish the low speed gear stage, and in order to provide a region in which a hydraulic pressure, which is needed to engage the rotational element when the low speed gear stage is established, is lower than a hydraulic pressure, which is needed to engage the rotational element when the rotational element is stopped from rotating in the direction of the de-energizing operation, the region being an operational range such that the band brake is set so as to include the operational range; and
  - the control unit includes a disengaging device which disengages the frictional engagement element at a downshift from the high speed gear stage to the low speed gear stage, a synchronization determining device which determines that the input rotation of the automatic transmission is synchronized with the rotation at the low speed gear stage, a constant pressure maintaining device which maintains the hydraulic pressure applied to the hydraulic servo of the band brake with a waiting pressure, which is lower by a predetermined amount than a pressure stopping the rotation of the rotational element in the direction of the self-energizing operation and with which the input rotation is raced by a basic racing amount after a synchronizing point, until the synchronization is determined by the synchronization determining device, and a pressure increasing device which increases the hydraulic pressure applied to the hydraulic servo from the waiting pressure in order to stop the rotational element from rotating after determining the synchronization.

2. The control system for an automatic transmission according to claim 1, wherein the pressure increasing device increases the hydraulic pressure with a predetermined rate, which increases when a throttle opening of an engine increases, after determining the synchronization.

3. The control system for an automatic transmission according to claim 1, wherein the pressure increasing device increases the hydraulic pressure at a predetermined rate, and controls the hydraulic pressure in order that the input rotation speed changes at a target rate which reduces when the output rotation speed of the automatic transmission reduces.

4. The control system for an automatic transmission according to claim 1, wherein the control unit stores a premeditated waiting pressure value for the basic racing amount to occur, and includes a maximum racing amount detecting device that detects a maximum racing amount of the input rotation speed based on the input rotation speed of the automatic transmission, and further includes a learning device which renews the stored waiting pressure value in order to achieve the basic racing amount by comparing the basic racing amount with the maximum racing amount after the end of the downshift.

5. The control system for an automatic transmission according to claim 4, wherein the pressure increasing device increases the hydraulic pressure with a predetermined rate, which increases when the throttle opening of the engine increases, after determining the synchronization.

6. The control system for an automatic transmission according to claim 4, wherein the pressure increasing device increases the hydraulic pressure at a predetermined rate and control the hydraulic pressure in order that the input rotation speed changes at a target rate which reduces when the output rotation speed of the automatic transmission reduces.

7. The control system for an automatic transmission according to claim 4, wherein multiple waiting pressure values are stored in the control unit, and correspond to a throttle opening of an engine and an output rotation speed as parameters, and the waiting pressure value is selected corresponding to the throttle opening and the output rotation speed at a shift start.

8. The control system for an automatic transmission according to claim 7, wherein the pressure increasing device increases the hydraulic pressure with a predetermined rate, which increases when the throttle opening of the engine increases, after determining the synchronization.

9. The control system for an automatic transmission according to claim 7, wherein the pressure increasing device increases the hydraulic pressure at a predetermined rate, and controls the hydraulic pressure in order that the input rotation speed changes at a target rate which reduces when the output rotation speed of the automatic transmission reduces.

10. The control system for an automatic transmission according to claim 1, wherein multiple waiting pressure values are stored in the control unit, and correspond to a throttle opening of an engine and an output rotation speed as parameters, and the waiting pressure value is selected corresponding to the throttle opening and the output rotation speed at a shift start.

11. The control system for an automatic transmission according to claim 10, wherein the pressure increasing device increases the hydraulic pressure with a predetermined rate, which increases when the throttle opening of the engine increases, after determining the synchronization.

12. The control system for an automatic transmission according to claim 10, wherein the pressure increasing device increases the hydraulic pressure at a predetermined rate, and controls the hydraulic pressure in order that the input rotation speed changes at a target rate which reduces when the output rotation speed of the automatic transmission reduces.

13. A method of controlling an automatic transmission that includes:

a frictional engagement element engageable to establish a high speed gear stage; and

a rotational element that can be stopped from rotating with a brake to establish a low speed gear stage, a rotational direction of the rotational element when the high speed gear stage has been established being opposite to a direction of torque to the rotational element created when the rotational element is stopped from rotating to establish the low speed gear stage;

wherein the brake is formed of a band brake which has a difference of engagement force between a self-energizing operation and a de-energizing operation, and includes a hydraulic servo;

the band brake is set so that a direction of the self-energizing operation is the same as the direction of the torque to the rotational element created when the rotational element is stopped from rotating to establish the low speed gear stage, and in order to provide a region in which a hydraulic pressure, which is needed to engage the rotational element when the low speed gear stage is established, is lower than a hydraulic pressure, which is needed to engage the rotational element when the rotational element is stopped from rotating in the direction of the de-energizing operation, the region being an operational range such that the band brake is set so as to include the operational range; the method comprising the steps of:

controlling hydraulic pressures applied to the frictional engagement element and the brake;

disengaging the frictional engagement element at a downshift from the high speed gear stage to the low speed gear stage with a disengaging device;

determining that the input rotation of the automatic transmission is synchronized with the rotation at the low speed gear stage with a synchronization determining device;

maintaining with a constant pressure maintaining device the hydraulic pressure applied to the hydraulic servo of the band brake with a waiting pressure, which is lower by a predetermined amount than a pressure stopping the rotation of the rotational element in the direction of the self-energizing operation and with which the input rotation is raced by a basic racing amount after a synchronizing point, until the synchronization is determined by the synchronization determining device; and

increasing with a pressure increasing device the hydraulic pressure applied to the hydraulic servo from the waiting pressure in order to stop the rotational element from rotating after determining the synchronization.

14. The method according to claim 13, further including the steps of:

controlling a premeditated waiting pressure value for the basic racing amount to occur;

detecting with a maximum racing amount detecting device a maximum racing amount of the input rotation speed based on the input rotation speed of the automatic transmission; and

renewing with a learning device the stored waiting pressure value in order to achieve the basic racing amount by comparing the basic racing amount with the maximum racing amount after the end of the downshift.

15. A control system for an automatic transmission that has a high speed gear stage, a low speed gear stage, and an input rotation, comprising:

a frictional engagement element engageable to establish the high speed gear stage;

a rotational element that is stopped from rotating with a brake to establish the low speed gear stage, a rotational direction of the rotational element when the high speed gear stage has been established being opposite to a direction of torque to the rotational element created when the rotational element is stopped from rotating to establish the low speed gear stage; and

a control unit that controls hydraulic pressures applied to the frictional engagement element and the brake, wherein:

the brake is formed of a band brake which has a difference of engagement force between a self-energizing operation and a de-energizing operation, and includes a hydraulic servo;

the band brake is set so that a direction of the self-energizing operation is the same as the direction of the torque to the rotational element created when the rotational element is stopped from rotating to establish the low speed gear stage, and in order to provide a region in which a hydraulic pressure, which is needed to engage the rotational element when the low speed gear stage is established, is lower than a hydraulic pressure, which is needed to engage the rotational element when the rotational element is stopped from rotating in the direction of the de-energizing operation, the region being an operational range such that the band brake is set so as to include the operational range; and

the control unit includes disengaging means for disengaging the frictional engagement element at a downshift from the high speed gear stage to the low speed gear stage, synchronization determining means for determining that the input rotation of the automatic transmission is synchronized with the rotation at the low speed gear stage, constant pressure maintaining means for maintaining the hydraulic pressure applied to the hydraulic servo of the band brake with a waiting pressure, which is lower by a predetermined amount than a pressure stopping the rotation of the rotational element in the direction of the self-energizing operation and with which the input rotation is raced by a basic racing amount after a synchronizing point, until the synchronization is determined by the synchronization determining

means, and pressure increasing means for increasing the hydraulic pressure applied to the hydraulic servo from the waiting pressure in order to stop the rotational element from rotating after determining the synchronization.

16. The control system for an automatic transmission according to claim 15, wherein the control unit stores a premeditated waiting pressure value for the basic racing amount to occur, and includes maximum racing amount detecting means for detecting a maximum racing amount of the input rotation speed based on the input rotation speed of the automatic transmission, and further includes learning means for renewing the stored waiting pressure value in order to achieve the basic racing amount by comparing the basic racing amount with the maximum racing amount after the end of the downshift.

* * * * *

United States Patent [19]

Hatta et al.

[11] **Patent Number: 5,003,842**

[45] **Date of Patent: Apr. 2, 1991**

[54] **CONTROL DEVICE FOR AUTOMATIC TRANSMISSION GEAR SYSTEM**

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[73] Assignee: **Mitsubishi Jidoshal Kogyo Kabushiki Kaisha**, Tokyo, Japan

[21] Appl. No.: **590,601**

[22] Filed: **Sep. 25, 1990**

Related U.S. Application Data

[63] Continuation of Ser. No. 463,935, Jan. 8, 1990, abandoned, which is a continuation of Ser. No. 267,667, Nov. 4, 1988, abandoned, which is a continuation of Ser. No. 81,586, Aug. 3, 1987, abandoned, which is a continuation of Ser. No. 912,686, Sep. 26, 1986, abandoned, which is a continuation of Ser. No. 530,642, Sep. 9, 1983, abandoned.

[30] **Foreign Application Priority Data**

Sep. 10, 1982 [JP] Japan 157644

[51] **Int. Cl.**[5] **B60K 41/06**

[52] **U.S. Cl.** **74/867**; 188/1.11

[58] **Field of Search** 74/866, 867, 868, 869; 188/1.11

[56] **References Cited**

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Primary Examiner—Dwight Diehl

[57] **ABSTRACT**

In an automatic transmission gear system which selectively actuates a plurality of frictional elements by oil pressure such as clutches and brakes that are provided in the transmission gear system in order to obtain different gear ratios, one frictional element is disengaged concurrently with engaging of another frictional element for switching the gear ratio (speed change). At that time, it is important to adjust the timing for actuating the frictional elements depending on the conditions of the engine and the vehicle itself. The present invention relates to a control device for an automatic transmission gear system which assures smooth gear shifting at all times by adequately controlling the overlap of the torque capacities of the frictional elements.

5 Claims, 4 Drawing Sheets

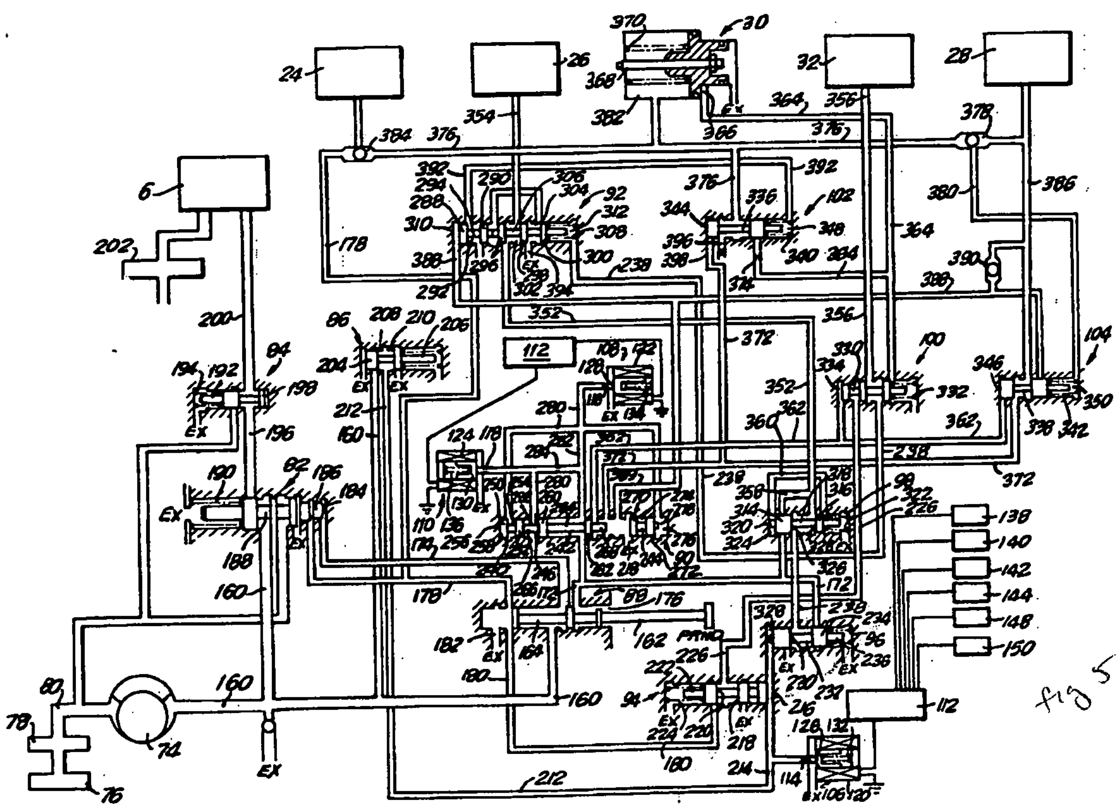

fig 5

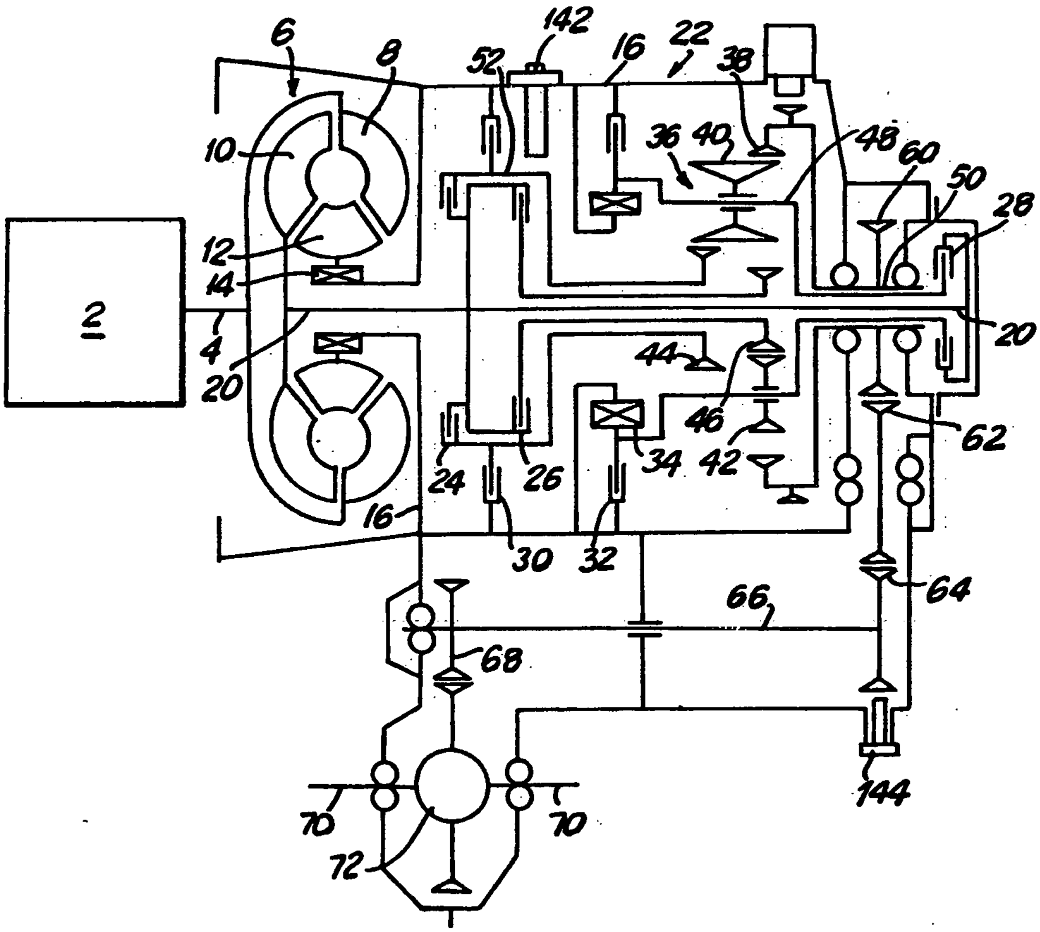

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FIG. 1

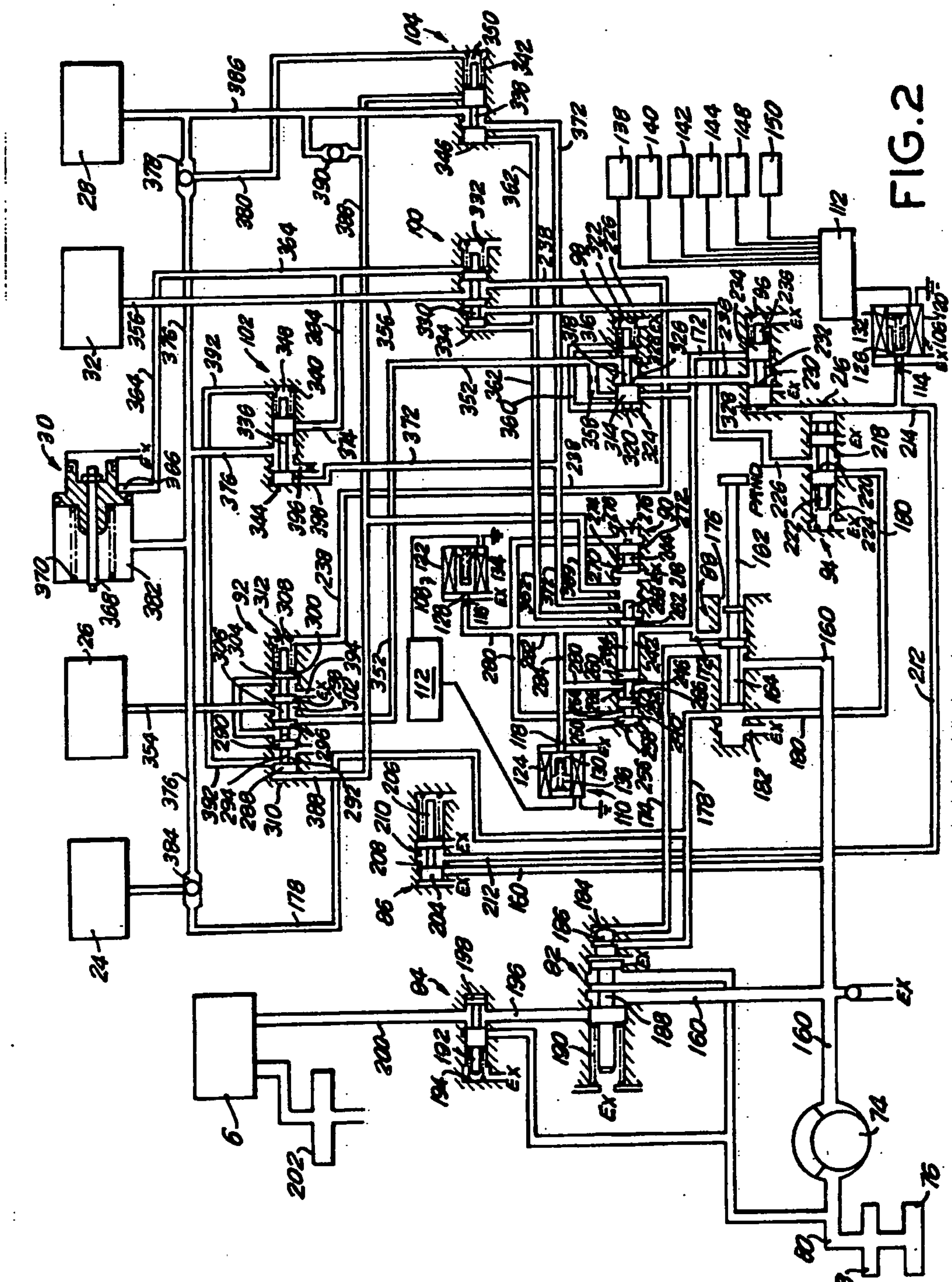


FIG. 2

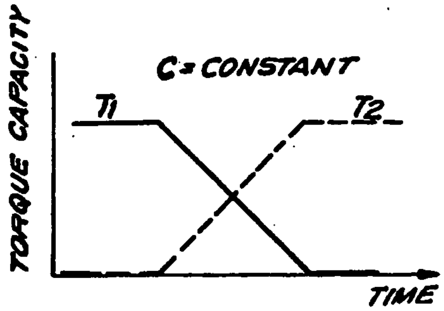


FIG.3(a)

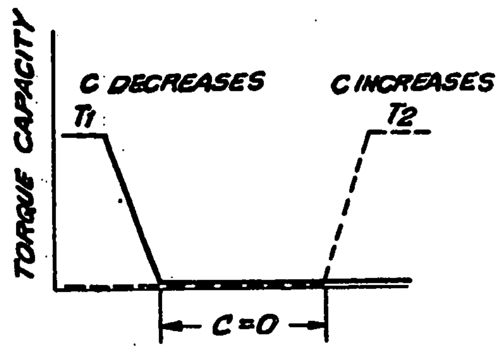


FIG.3(b)

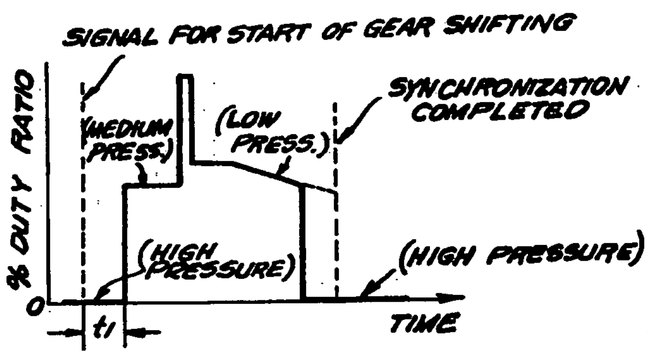
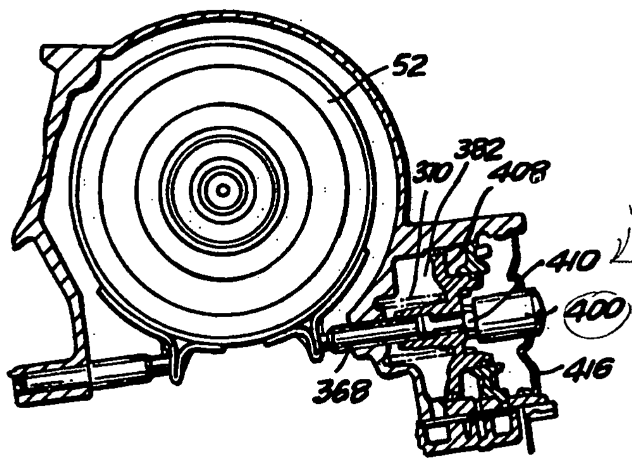


FIG.4



magnet positioning

stroke detecting means

FIG.5

* move piston 62, move rod 68 sense position of rod, can also sense piston @ least when moving toward lock band.

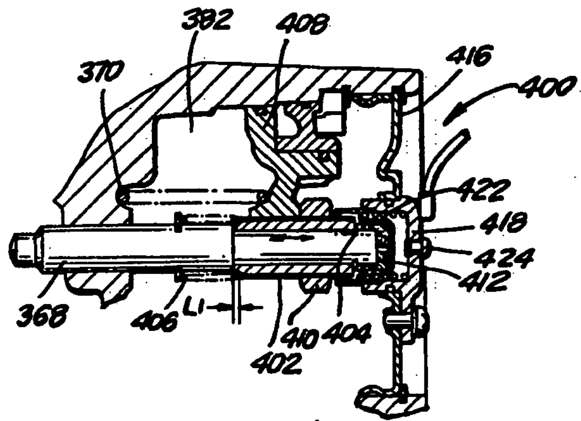


FIG.6(a)

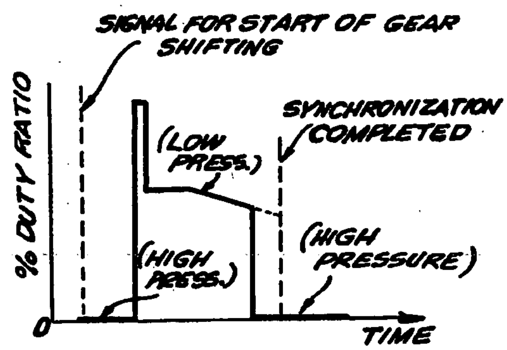


FIG.7

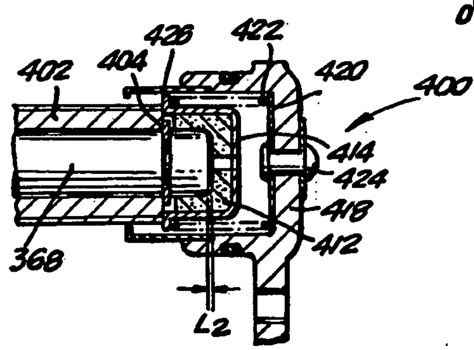


FIG.6(b)

CONTROL DEVICE FOR AUTOMATIC TRANSMISSION GEAR SYSTEM

This application is a continuation f application Ser. No. 463,935, filed Jan. 8, 1990; which is a continuation of application Ser. N . 07/267,667 filed Nov. 4, 1988; which is a continuation of application Ser. No. 07/081,586 filed Aug. 3, 1987; which is a continuation of application Ser. No. 06/912,686 filed Sept. 26, 1986; which is a continuation of application Ser. No. 06/530,642 filed Sept. 9, 1983.

BACKGROUND OF THE INVENTION

The present invention relates to an improvement of the control device for an automatic transmission gear system and aims to alleviate the shock caused by gear shifting.

Automatic transmission gear systems used in automobiles are generally capable of selectively actuating a plurality of frictional elements such as clutches and brakes that are provided in the transmission gear system by oil pressure in order to obtain different gear ratios, and for effecting the gear shift, one of the frictional elements is disengaged while another one is engaged.

In an automatic transmission gear system of the above construction, adequate control of timing for disengaging one of the frictional elements and engaging another element is of great importance in alleviating shock due to speed changes and in assuring comfortable driving as well as preventing any damage to the transmission gear system.

For example, if the disengagement occurs too early and the engagement too late, damage occurs during the time between the disengagement and the engagement. On the other hand, if the disengagement occurs too late and the engagement too early, a state of both frictional elements being engaged occurs. This leads to contradictions in a speed change gear assembly and causes damage to the vehicle as a whole as well as putting an excessive load on the frictional elements and the gear systems, thus inducing damage thereto.

Therefore, when the gear ratio of the transmission is changed from lower to higher speed with the engine in a driving state, an overlap of torque capacities of the frictional elements proportional to the output torque of the engine is desirably provided between the disengagement and engagement to prevent the engine from racing; while on the other hand, when the engine is in the driven state, it is preferable to provide a suitable time interval after the frictional element for lower speed driving is released until the rotational speed of the engine decreases enough to accommodate the driving condition at a higher speed. In this way, the frictional elements for higher speed driving can be engaged (the interval is defined as a zero overlap or an overlap of negligible amount).

On the contrary, when the gear ratio of the transmission is changed from higher to lower speed with the engine in a driving state, it is desirable that the timing for disengaging and engaging the frictional elements be controlled corresponding to the driving speed of the vehicle. When the vehicle is running at a higher speed, it is preferabl that, after disengaging the frictional element for higher speed driving, a small amount of said overlap be provided to let the rotational speed of the engine increase enough to accommodate the driving condition at a lower speed, before the frictional element for lower speed driving is engaged. As the vehicle slows down, it is necessary to increase the amount of the overlap. Especially at a low speed, it is desirable that the interval is reduced to almost zero, and the verlap increased.

Thus, it is necessary to provide the overlap at lower speeds and reducing it gradually as the vehicle gathers speed. However, if the overlap of the engagement is small when the engine is in a driven state, an interval takes place causing the engine braking effect of the vehicle to be cancelled for a while, and thereafter the frictional element for lower speed driving is engaged causing the vehicle to come under forcible engine braking effect, resulting in impaired comfort of driving.

A one-directional brake instead of ordinary brakes can be used to automatically change such gear ratios, but this cannot be adapted to all of the frictional elements in the transmission. Also, a control device activated by oil pressure can be used which detects the change in the rotational speed of axis of the transmission gear system so that the oil pressure can be supplied without delay to the frictional elements for coupling when synchronization is attained (Jap. Pat. Pub. Sho 54-35631).

SUMMARY OF THE INVENTION

The present invention aims to provide a control device for an automatic transmission gear system which is capable of alleviating the shock at the time of shifting from one gear ratio to another by providing a suitable overlap of the frictional elements dependent on the driving condition of a vehicle when such elements are activated to change the gear ratio.

Another object of the present invention is to provide a control device for an automatic transmission gear system which can reduce the time needed for gear shifting without causing shock.

Still another object of the present invention is to provide a control device for an automatic transmission gear system which can be adapted to a wider range of speeds.

In an automatic transmission gear system which is provided with a speed change gear assembly having plural gear ratios between input and output shafts, a control device according to the present invention to achieve the above objects comprises: a frictional element actuated by oil pressure to obtain a gear ratio for lower speed driving; a frictional element actuated by oil pressure to obtain a gear ratio for higher speed driving; a synchronization detecting means to detect whether said gear ratios have been obtained or not; a switching valve for switching oil passages leading to the frictional elements at a time of shifting between the lower and the higher speed driving; an overlap regulating means to provide an overlap of torque capacities of the frictional elements when gear shifting between the higher and the lower speed; and an electronic control device which controls the oil pressure to be supplied to said overlap regulating means to control the amount of overlap in such a way that, during gear shifting from the higher to the lower speed driving, said amount of overlap is controlled to decrease when engagement of the higher-speed gear ratio is released and that amount of overlap is controlled to increase when engagement of the lower-speed gear ratio is obtained.

BRIEF DESCRIPTION OF THE DRAWINGS

FIGS. 1 and 2 show an automatic transmission gear system to which the device according to the present invention can be adapted. FIG. 1 shows the construc-

tion of the power transmission means in schematic view. FIG. 2 shows the construction of the oil pressure control means. FIGS. 3(*a*) and 3(*b*) are graphs explaining the amount f overlap respectively. FIGS. 4 and 7 are graphs explaining oil pressure control at the time of speed changes. FIGS. 5, 6(*a*) and 6(*b*) show the stroke detection means, with FIG. 5 showing the same in cross section and FIGS. 6(*a*) and 6(*b*) showing the essential parts in enlarged sectional views respectively.

DETAILED DESCRIPTION OF THE EMBODIMENT

The embodiments of the present invention will now be described in detail with reference to the accompanying drawings.

Referring first to FIG. 1 which shows the construction of an automatic transmission gear system to which the control device according to the present invention is to be adapted, an engine 2 which constitutes the power source for driving the vehicle is provided with a crank shaft 4 which is directly connected to a pump 8 of a torque converter 6. The torque converter 6 comprises the pump 8, a turbine 10, a starter 12, and a one-way clutch 14, and the starter 12 is connected to a case 16 via the one-way clutch 14. The stater 12 is so constructed that it can be rotated in the same direction as the crank shaft 4 but is not permitted to rotate in the reverse direction by means of said one-way clutch 14.

The torque transmitted to the turbine 10 is then transmitted via an input shaft 20 to a speed change gear assembly 22 which is provided behind said turbine 10 and which has four forward speeds and one reverse speed.

Said speed change gear assembly 22 comprises 3 clutches 24, 26 and 28; 2 brakes 30 and 32; one one-way clutch 34, and one ravigneraux type planetary gear set 36. Said planetary gear set 36 comprises a ring gear 38, a long pinion gear 40, a short pinion gear 42, a front sun gear 44, a rear sun gear 46 and a carrier 48 which rotatably supports said pinion gears 40 and 42 and also rotatable itself. The ring gear 38 is coupled to an output shaft 50 and the front sun gear 44 is coupled to the input shaft 20 by means of a kickdown drum 52 and a front clutch 24. The rear sun gear 46 is coupled to the input shaft 20 via a rear clutch 26. The carrier 48 is fixedly coupled to the case 16 via the low reverse brake 32 and the one-way clutch 34 that are arranged so that they are functionally parallel. The carrier 48 is also coupled to the input shaft 20 via the 4th speed clutch provided behind the speed change gear assembly 22. Said kickdown drum 52 can be fixedly coupled to the case 16 by means of the kickdown brake 30. The torque transmitted to the planetary gear set 36 is further transmitted from an output gear 60 fixed to the output shaft 50 to a driven gear 64 via an idle gear 62, and to a differential gear mechanism 72 via a transfer shaft 66 fixed to said driven gear 64 and a helical gear 68, the said differential gear mechanism 72 being coupled to a driving shaft 70.

Clutches and brakes mentioned above each comprise a frictional coupling means provided with a piston or servo means for engagement, and are actuated by oil pressure generated at an oil pump 74 which is shown in FIG. 2 and is driven by the engine 2 via the pump 8 of the torque converter 6. Said oil pressure is selectively distributed by means of an oil pressure control device to the clutches and the brakes according to the driving condition of the vehicle which is detected by various detecting means. Appropriate combinations of different clutches and brakes in operation will achieve 4 forward speeds and 1 reverse speed as sh wn in Table 1. In this Table, the symbol ○ denotes that the clutches or brakes are engaged, and the symbol ● denotes that the rotation of the carrier is locked by the function f the one-way clutch 34 immediately bef re the low reverse brake 34 is engaged for shifting operati n.

TABLE 1

	Gears					
Frictional Element	1st	2nd	3rd	4th	R	N.P.
Front Clutch 24		○	○		○	
Rear Clutch 26	○	○	○			
Kickdown Brake 30		○		○		
Low Reverse Brake 32	○				○	
One-way Clutch 34	●					
4th Speed Clutch 28				○		

The electronic oil pressure control device which is responsible for achieving the gear shiftings shown in Table 1 in the transmission gear system shown in FIG. 1 will now be described.

The oil pressure control device shown in FIG. 2 controls the oil pressure according to the driving condition of the vehicle. The oil pressure is supplied by an oil pump 74 from an oil sump 76 via an oil filter 78 and an oil passage 80 in order to actuate the piston or servo mechanism of the clutches 24, 26 28 and the brakes 30, 32 respectively, and to the torque converter 6. The control device comprises as the principal components a pressure regulating valve 82, torque converter control valve 84, a pressure reducing valve 86, a manual valve 88, a shift control valve 90, a rear clutch control valve 92, an N-R control valve 94, an oil pressure control valve 96 for gear shifting, an N-D control valve 98, a 1st-2nd speed shift valve 100, a 2nd-3rd speed and 4th-3rd speed shift valve 102, a 4th speed clutch control valve 104 and three electromagnetic valves 106, 108 and 110. These components are connected by means of the oil passages. The shift control valve 90, the 1st-2nd speed shift valve 100, the 2nd-3rd and 4th-3rd speed shift valve 102 and the 4th speed clutch control valve 104 function to switch the oil passages leading to respective frictional elements 24, 26, 28, 30 and 32 for changing the gear ratios.

Said electromagnetic valves 106, 108 and 110 are similar in structure and are of the type which controls the opening and the closing of orifices 114, 116 and 118 by means of electric signals transmitted from an electronic control device 112 and is closed when the electric signals are cut off. Each electromagnetic valve comprises coils 120, 122, 124 and valves 126, 128 and 130 respectively, which are provided inside said coils to switch ON/OFF orifices 114, 116 and 118, and springs 132, 134 and 136 which energize said valves.

The electronic control device 112 contains a means which detects driving state and accordingly decides which one of the electromagnetic valves 108 or 110 is to be closed and opened, and a means to detect the start of gear shifting. The said device 112 controls the ON/-OFF of duty control for the electromagnetic valve 106 and also the oil pressure by controlling the valve opening period within one cycle by varying the pulse-width of pulsating electric signal of 50 Hz supplied to the valve 106. The control device 112 further controls the opening/closing of the electromagnetic valves 108 and 110. The input elements of the said device 112 include a detecting means 138 which detects the opening degree

of a throttle valve (not shown) of the engine 2 or the negative pressure of the intake manifold, a detection means 140 which detects the rotational speed of the engine, a detecting means 142 which detects the rotational speed of kickdown drum 52 shown in FIG. 1, a detecting means 144 which detects th rotational speed of the driven gear 64 to thereby detect the rotational speed of the output shaft 50, a temperature detection means 146 for detecting the temperature of the lubricant oil, a position detecting means 148 which detects the position of the selector lever and a position detecting means 150 which detects the position of an auxiliary switch.

Pressurized oil discharge from said oil pump 74 is supplied to the oil pressure regulating valve 82, the manual valve 88 and the pressure reducing valve 84 via an oil passage 160.

The manual valve 88 is provided with 4 positions, i.e. D, N, R and P. When the D position is selected, the oil passage 160 connects with oil passages 172 and 174, whereby said speed change gear assembly 22 achieves gear shiftings of 4 forward speeds from the first to the fourth by the ON/OFF combinations of the electromagnetic valves 108 and 110 as shown in Table 2. At the N position, the oil passage 160 connects only with the oil passage 174 while the oil passage 172 is connected to an oil outlet port 176, whereby the speed change gear assembly 22 comes to a neutral state. At the R position, the oil passage 160 connects with oil passages 178 and 180 to allow the speed change gear assembly 22 to come into the reverse driving arrangement. And, at the P position, all the oil passages leading to the manual valve 88 are connected with the oil outlet port 176 or 182 to thereby bring the transmission gear 22 to practically a neutral state.

TABLE 2

Gears	Solenoid Valve 108	Solenoid Valve 110
1st	ON	ON
2nd	OFF	ON
3rd	OFF	OFF
4th	ON	OFF

The pressure regulating valve 82 comprises a spool 188 having pressure receiving surfaces 184 and 186, and a spring 190. As the oil pressure from the oil passage 160 acts on the pressure receiving surface 184 via the oil passage 174, oil pressure in the oil passage 160 is regulated to a predetermined value of 6 kg/cm^2 (hereinafter referred to as "the line pressure"). When the oil pressure from the oil passage 160 acts on the pressure receiving surface 186 via the oil passage 178, said oil pressure is regulated to 14.6 kg/cm^2.

The torque converter control valve 84, comprising a spool 192 and a spring 194, controls the pressurized oil supplied from the pressure regulating valve 82 via the oil passage 196 to be 2.5 kg/cm^2 by balancing the oil pressure acting on the right end surface (as viewed in the drawing) of the spool 192 with the urging force of the spring 194, and supplies the thus regulated oil pressure to the torque converter 6 via the oil passage 200. The oil discharged from the torque converter 6 is supplied to respective lubricating parts of the speed change gear assembly via an oil cooler 202.

The pressure reducing valve 86, comprising a spool 204 and a spring 206, reduces and regulates the oil pressure from the oil passage 160 to be at 2.4 kg/cm^2 by balancing the urging force of the spring 206 with the oil pressure caused by the area difference of the opposing pressure receiving surfaces 208 and 210 that are f rmed in the spool 204, and then supplies the oil pressure to the oil passage 212. Pressurized oil thus regulated (reduced) and supplied to the N-R control valve 94, the oil pressure control valve 96 and the orifice 114 of the electromagnetic valve 106 via the oil passage 212 and an orifice 214.

The N-R control valve 94, which comprises a spool 222 having pressure receiving surfaces 216, 218 and 220 and a spring 224, regulates the oil pressure of the oil passage 226 to a desired value by balancing the oil pressure acting on the surface 216 with the combined force of the oil pressure caused by the area difference of the surfaces 218 and 220 and the urging force of the spring 224.

The oil pressure control valve 96, which comprises a spool 234 having pressure receiving surfaces 228, 230 and 232 and a spring 236, regulates the oil pressure of the oil passage 238 to a desired value by balancing the oil pressure acting on the surface 228 with the combined force of the oil pressure caused by the area difference of the surfaces 230 and 232 and the urging force of the spring 236.

Said oil pressure thus regulated and supplied to the oil passage 226 controls the low reverse brake 32 at the time of shifting the gear to the reverse position. Oil pressure regulated and supplied to the oil passage 238 controls the front clutch 24, the rear clutch 26, the kickdown brake 30 and the low reverse brake 32 when the vehicle is in advance or is at a halt.

The electromagnetic valve 106 is subjected to be duty controlled by the electronic control device 112 by means of pulsating electric signal of 50 Hz whose pulse width is variable depending on the driving condition of the vehicle. By the change of the pulse width, the ON/-OFF time of the orifice 114 is controlled so that the oil pressure in the oil passage 212 which is at the downstream of the orifice 214, or in other words the oil pressure P_1 which acts on the pressure receiving surface 228 of the oil pressure control valve 96 and the surface 216 of the N-R control valve 94 can be controlled. By controlling said oil pressure P_1, oil pressure supplied to respective frictional elements are controlled and controls the engaging action of the respective frictional elements. Also, the oil pressure P_1 controls oil pressure supplied to an overlap regulating means which controls the amount of overlap. For example, suppose the diameter of orifice 214 is 0.8 mm and that of orifice 114 is 1.4 mm said oil pressure P_1 is regulated within the range of approximately from 0.3 to 2.1 kg/cm^2. Accordingly, the regulated oil pressure appearing in oil passages 226 and 238 will vary in proportion to the increase or decrease of said oil pressure P_1 within the range of approximately from 0 kg/cm^2 to the supplied oil pressure (oil pressure in oil passage 180 or 172).

The timing to start operating the electromagnetic valve 106 and the duration time is determined by the signals transmitted from the detecting means 138 for the engine load, sensors 140, 142 and 144 which detect the rotational speed as well as signals from the gear shift detecting means which detects the start of gear shifting, and the detecting means comprising two sensors 142 and 144 that are contained in the electronic control device 112 to detect the engagement timing.

The shift control valve 90 is controlled by various ON/OFF combinati ns of the electromagnetic valves

108 and 110, and comprises three spools 240, 242 and 244 and two stoppers 246 and 248, wherein said spool 240 is provided with lands 250 and 252, an annular groove 254 and the oil passage 258 which communicates with an oil chamber 256 located at the left of the land 254 and the annular groov 254 as viewed in the drawing; said spool 242 is provided with lands 260 and 262, each having different diameters, an annular groove 264 and pressing members 266 and 268 which are abut against the spools 240 and 244 respectively; and the spool 244 is provided with lands 270 and 272, an annular groove 274 and an oil passage 278 which communicates with an oil chamber 276 located at the right of the land 272 and the annular groove 274. The stopper 246 is interposed between the spools 240 and 242 and fixed to the casing; the stopper 248 is interposed between the spools 242 and 244 and fixed to the casing. The oil passage 172 connected with an oil passage 280 at all times via the annular groove 264, and said oil passage 280 communicates with the orifice 116, the oil chamber 256 located at the left and the oil chamber 276 at the right via an orifice 282 and also with the orifice 118 and an oil chamber 286 located in between the spools 240 and 242 via an orifice 284.

The rear clutch control valve 92 comprises a spool 294 provided with a land 288 and a land 290 which is smaller in diameter than the land 288 and also an annular groove 292, a spool 306 provided with three lands 296, 298 and 300 which are the same in diameter as the land 288 and annular grooves 302 and 304, and a spring 308. When the pressing force of the oil pressure introduced in an oil chamber 310 at the left as shown in FIG. 2, said pressing force acting on the pressure receiving end surface of the land 288 becomes greater than the combined force of the oil pressure introduced in an oil chamber 312 at the right as shown in FIG. 2 and the pressing force acting on the pressure receiving end surface of the land 300 and the urging force of the spring 308 causes the spools 294 and 306 to move toward the right end as in the drawing. When the spools are situated at the right end, the oil pressure acts in between the lands 290 and 296, so that the spool 294 al ne moves toward the left end as the oil pressure in the oil chamber 310 is released. When the oil pressure acting on the left end of the pressure receiving surface 296 becomes less than the combined force of the oil pressure in said oil chamber 312 and the urging force of the spring 308, it then causes the spool 306 to move toward the left.

The N-D control valve 98 which comprises a spool 320 provided with lands 314, 316 and an annular groove 318, and a spring 322, selectively switches the position of the spool 320 between the left end as shown in FIG. 2 and the right end (not shown) of the valve 98 in accordance with the direction of the combined forces of the oil pressure acting on the pressure receiving surfaces of the spool 320 with the urging force of a spring 322.

The 1st-2nd speed shift valve 100 which comprises a spool 330 and a spring 332, switches the position of said spool 330 between the left end as indicated in FIG. 2 and the right end (not indicated) by suitably supplying or releasing the line pressure which acts on a pressure receiving surface 334 of the left end of the spool 330.

When the line pressure is so supplied to act on the surface 334, the spool 330 is caused to move toward the right by the line pressure, whereas when the line pressure is exhausted, it will be pressed toward the left end by the force of the spring 332.

The 2nd-3rd and 4th-3rd speed shift valve 102 and the 4th speed clutch control valve 104 also comprise spools 336 and 338 and springs 340 and 342 respectively. At the left of the respective spools 336 and 338 are the oil pressure chambers 344 and 346 to which the line pressure is introduced and at the right are the il chambers 348 and 350 respectively. Thus, the respective spools will be selectively switched between the left end as indicated in FIG. 2 and the right end not indicated.

Now the functions of the oil pressure control device will be described in conjunction with the gear shift control. As for the conventional gear shift control, details thereof will be omitted as it is well known in the art (Jap. Pat. Appln. No. Sho 56-144237).

First, the concept of overlap which is of critical significance in the present invention will be explained.

Suppose that the torque capacity of the frictional element which determines the lower-speed gear ratio is expressed as T_1, that of the frictional element which determines the higher-speed gear ratio as T_2, with constants a_1, a_2 such as the piston area of each frictional element, the number of clutch plates, radius of clutches and coefficient of friction, then the amount of overlap C can be expressed by the following equation:

$$C = a_1 T_1 + a_2 T_2$$

Thus, by varying the oil pressure supplied to the respective frictional elements, the value of T_1 or T_2 can be varied to thereby change the amount of overlap C. FIG. 3(*a*) shows a case where the shift takes place with the amount of overlap C at a constant level, whereas FIG. 3(*b*) shows a case where the shift takes place with the amount of overlap C varied.

The first embodiment of the present invention will be described with respect to the 3rd-2nd speed downshift. Referring again to FIG. 2, an oil chamber 382 of the kickdown brake 30 (the frictional element for obtaining the lower-speed gear ratio) is connected with the oil pressure chamber of the front clutch 24 (the frictional element for determining the higher-speed gear ratio) via an oil passage 376 thus constitutes the overlap regulating means. In order to disengage the 3rd speed, therefore, it is necessary to exhaust the pressurized oil supplied to the front clutch. For this, it is necessary to lower the pressure of oil to be supplied to an oil chamber 366 of the kickdown brake 30 and reduce the oil to be pumped out from an oil chamber 382 of said brake 30, and at the same time the 2nd-3rd speed and 4th-3rd speed shift valve 102 must be moved toward the 2nd speed position (at the left end as shown in FIG. 2). When, however, the oil supplied to the chamber 366 of the kickdown brake 30 is maintained at a lower pressure, the kickdown brake 30 is not ready for engagement and even if the pressure is elevated as soon as synchronization of the 2nd speed is completed, the kickdown brake 30 will not be engaged in time. Thus, it becomes necessary to supply high pressure oil until the brake piston moves and preparation for engaging is completed. But, if high pressure oil is continuously supplied, the engaging force becomes great so rapidly at the engagement of the kickdown brake 30 and the front clutch 24 disengaging at the same time resulting shock.

Now, as the output signal indicating the start of gear shifting from 3rd to 2nd speed is transmitted fr m the electronic control device 112 according to the various input signals which represent the driving condition of the vehicle, the electromagnetic valve 108 will be de-

energized whil the electromagnetic valve 110 will be energized. Because the oil pressure in the oil passag 372 is exhausted by the function of the shift control valve 90, the spool 336 of the 2nd-3rd and 4th-3rd speed shift valve 102 is moved toward the left. Then the oil passage 376 communicates with an exhaust port 396, causing the oil pressure in an oil chamber 350 of the 4th speed clutch control valve 104 to exhaust via a check valve 378 and the oil passage 380. So, the spool 338 of said 4th speed clutch control valve 104 moves to the right, causing the oil pressure of the clutch 28 to exhaust via the oil passage 386, 388 and the clutch 28 is disengaged. At the same time the duty ratio of the electromagnetic valve 106 is controlled by the electronic control device 112 as the signal indicating the start of gear shifting is transmitted. The control is carried out in such a way that the oil pressure to be supplied from the oil pressure control valve 96 to the oil chamber 336 of the kickdown brake 30 via the lst-2nd speed shift valve 100 and the oil passage 364, be supplied at a higher pressure (line pressure) for a predetermined time t_1 until the brake piston will have been moved and ready for engaging, said time t_1 being predetermined from the supply oil pressure and the volume of the oil chamber 366 of the kickdown brake 30. This causes the piston in the kickdown brake 30 to move rapidly, and the oil in the oil chamber 382 is rapidly exhausted to the oil passage 376. Since exhaustion of the oil in the front clutch 24 is delayed, this will maintain the torque capacity of said clutch 24 and hold the overlap c to a large amount. After the predetermined time t_1 elapses, the duty ratio for the electromagnetic valve 116 is increased to obtain a medium pressure while maintaining the kickdown brake 30 ready for full engagement. As a result, the oil pressure in the front clutch 24 can be exhausted without being impaired by the exhaust oil from the oil chamber 382 of the kickdown brake 30 to thereby cause the 3rd speed gear to be disengaged. This disengagement of the 3rd-speed gear is detected by comparing the signals from the detecting means 142 which detects the rotational speed of the kickdown drum 52 and the detecting means 144 which detects the rotational speed of driven gear 64. Said detecting means 142, 144 function as a part of the synchronization detecting means. By means of the signal which indicates the disengagement of the 3rd speed gear, the oil pressure to be supplied to the kickdown brake 30 will be promptly reduced to thereby decrease the torque capacity of the front clutch 24 and thus the amount of overlap C, so that the change speed transmission assembly 22 will practically be in the neutral state in which the amount of overlap C is reduced to 0 (a state wherein the torque capacity of the two frictional elements 24 and 30 respectively becomes 0 to permit a time interval). Synchronization of the 2nd speed is detected by the signals from said rotational speed detecting means 142 and 144. (In this case, synchronization of the 2nd speed can be detected by detecting the time when the rotational speed of the kickdown drum 52 becomes zero.) By the signal indicating the synchronization of the 2nd speed, the oil pressure to be supplied to the kickdown brake 30 is promptly switched from lower to higher pressure, and the amount of overlap C is increased from 0 to a higher value by increasing the il pressure for engaging th kickdown brake 30 which is the frictional element for the 2nd speed. Thus, the gear shifting from the 3rd to the 2nd speed is completed. It is to be n ted that synchronization is deemed complete when approximately 300 rpm before synchronization is obtained considering th delay in increasing the low oil pressure to a higher value after the synchronization of the 2nd speed is detected.

As has been mentioned above, by regulating the amount of overlap C of the kickdown brake 30 and the front clutch 24 by controlling the duty ratio of th electromagnetic valve 106 by means of the electronic control device 112, the kickdown brake 30 will be smoothly engaged without shock, whereby the gear shifting from 3rd to 2nd speed is achieved.

In the above embodiment, the oil pressure is varied high→medium→low→high, but it is possible to vary the pressure high→low→high, skipping the medium range However, the time required for gear shifting can be shortened by providing the medium pressure range. This applies not only to the downshift (gear change) from 3rd to 2nd speed, but also to the downshift from the 4th to 3rd speed, in which case the engagement-/disengagement of the kickdown brake 30 and the front clutch is reversed.

The second embodiment of the present invention will now be explained with respect to the downshift from the 3rd to 2nd speed. The embodiment is almost identical in construction with the first embodiment except that a stroke detecting means 400 is provided to detect the stroke position of an oil pressure piston rod 368 in order to directly detect whether the kickdown brake 30 is ready for engagement.

The stroke detecting means 400 is attached to the base of the rod 368, as shown in FIGS. 5, 6(*a*) and 6(*b*), to which the oil pressure piston of the kickdown brake 30 is fixed. A stepped portion is formed at the base of the rod 368, and a sleeve 402 which is made of a conductive material such as stainless steel and formed about 1 mm shorter than the stepped portion, is slidably mounted on the outer circumference of the stepped portion and retained to the rod 368 by means of a retaining ring 404 in such a manner that it will have a stroke L_1 of this 1 mm difference. The sleeve is also urged toward the right in FIG. 6(*a*) by means of a spring 406. An oil pressure piston 408 is fixedly screwed to the threaded portion at the outer circumference of the sleeve 402 by means of a nut 410, and the spring 370 is interposed between said oil pressure piston 408 and the wall of the oil chamber 382. On the other hand, an insulating cap 412 made of an insulating material, such as resin is attached to the base of the rod 368. A spring 422 is interposed between a stopper 414 made of stainless steel which caps on the insulating caps 412 and a stainless washer 420 attached to the inner side of a resin mold cap 418 which attached to a cover 416. The arrangement is such that when the sleeve 402 is pressed toward the right by means of the spring 406 as shown in FIG. 6(*b*), there is formed gap L_2 of approximately 0.1 to 0.4 mm between the rod 368 and the insulating cap 412. When a terminal 424 provided at the center of the cap 418 is connected to the power source under the condition as shown in FIG. 6(*b*), an electric circuit connecting the terminal 424, the washer 420, the spring 422, the stopper 414, a contact point 426, the sleeve 402, the oil pressure piston 408 and the spring 370 is formed if the oil pressure piston is at its normal position. On the other hand, if the oil pressure piston 408 is pressed by the pressurized oil supplied to the oil chamber 366 at, the rod 368 is caused to shift its position without contracting the spring 406 which presses the sleeve 402 until just before the engagement of the kickdown brake 30. As the resistance against the movement of the rod

368 increases just before engagement, the oil pressure piston 408 inevitably moves for a distance of stroke L_1 while causing the spring 406 to contract As a result, the stopper 414 is permitted to move only for a distance of the gap L_2 because of the insulating cap 412 at the contact point 426 of the stopper 414 and the sleeve 402. This causes the contact point 426 to be left open, and the electric circuit will not be completed.

Thus, according to the present invention, the position of the kickdown brake 30 just before the engagement can be electrically detected. It should be noted that the gap L_2 can be formed between the retaining ring 414 and the insulating cap 412.

So, when a signal for the gear shifting from 3rd to 2nd speed, based on the signals representing the driving condition of the vehicle, is transmitted from the electronic control device 112, the electromagnetic valve 108 is subsequently de-energized and the electromagnetic valve 110 is energized, whereby the oil pressure in the oil passage 372 is exhausted by the function of the shift control valve 90. As a result, the spool 336 of the 2nd-3rd and 4th-3rd speed shift valve 102 is moved toward the left in FIG. 2. Then the oil passage 376 communicates with the exhaust port 396, causing the oil pressure in the oil chamber 350 of the 4th speed clutch control valve 104 to exhaust via the check valve 378 and the oil passage 380. So, the spool 338 of said 4th speed clutch control valve 104 moves to the right, causing the oil pressure of the clutch 28 to exhaust via the oil passage 386, 388, and the clutch 28 is disengaged. At the same time, the duty ratio of the electromagnetic valve 106, is controlled by the electronic control device 112 as the signal indicating the start of gear shifting is transmitted as indicated in FIG. 7, so that the oil pressure to be supplied from the oil pressure control valve 96 to the oil chamber 366 of the kickdown brake 30 via the oil passage 238, the 1st-2nd speed shift valve 100 and the oil passage 364 is first maintained high. This causes the piston in the kick-down brake 30 to move rapidly, and the oil in the oil chamber 382 is rapidly exhausted to the oil passage 376. Since exhaustion of the oil in the front clutch 24 is delayed, this will maintain the torque capacity of said clutch 24 and hold the overlap c to a large amount. As soon as the oil pressure piston 408 of the kickdown brake 30 shifts itself to the position just before engagement and the OFF signal is transmitted from the stroke detecting means 400, the oil pressure is reduced to decrease the amount of overlap C. Then the pressurized oil of the front clutch 24 is exhausted from the oil passage 376 under this condition to disengage the 3rd speed. Subsequently, the neutral state with the amount of overlap C=0 will follow before the complete synchronization of the 2nd speed is detected by the signals from the rotational speed detecting means 142 and 144 which also function as a part of the synchronization detecting means. By the signal indicating the synchronization of the 2nd speed, the low pressure oil which is supplied to the kick-down brake 30 to be promptly elevated to thereby increase the amount of overlap C. This permits the kickdown brake 30 to be engaged, and the gear shifting from 3rd to 2nd speed is completed.

It should be noted that considering the time delay actually to increase the oil pressure after the synchronization of the 2nd speed is detected, synchronization is deemed complete when approximately 300 rpm before synchronization is obtained.

In this second embodiment, th electronic control device 112 controls the duty ratio f the electromagnetic valve 106 in such a way that high oil pressure is supplied to the chamber 366 of the kickdown brake 30 until just bef re said brake 30 is engaged and that as soon as an OFF signal is transmitted from the stroke detecting means 400, the pressure f the oil to be supplied to the kickd wn brake 30 is lowered to thereby decrease the amount of overlap C. Upon completion of synchronization of the 2nd speed, the oil pressure supplied to the kickdown brake 30 is promptly elevated to thereby increase the amount of overlap C and to regulate the torque capacity to correspond with that of the frictional element with the gear ratio for 2nd speed driving. Thus, the second embodiment can reduce the time required before the start of effective gear shifting when shifting from 3rd to 2nd speed, compared to the above-mentioned first embodiment, and still enables smooth gear shifting without causing shock. The second embodiment also enables to shift down again from 3rd to 2nd speed during the gear shifting up from 2nd to 3rd speed, even if the position of the piston provided in the kickdown brake 30 is unsettled, whereas in the first embodiment one must wait until a predetermined time t_1 elapses for effective shifting down.

The third embodiment of the present invention will now be explained. This embodiment is almost identical in construction with the second embodiment except that the control mechanism of the electronic control device 112 is different. This embodiment is effective for engaging the kickdown brake 30 at the time of gear shifting from 1st to 2nd or 3rd to 4th speed.

The embodiment will now be explained in detail with respect to the gear shifting from 1st to 2nd speed in the automatic transmission system as shown in FIGS. 1 and 2.

When the accelerator pedal is stepped on under 1st gear driving and the vehicle gathers speed, a command signal to achieve 2nd speed is transmitted from the electronic control device 112 in response to the signals from the sensor 138 which detects the opening angle of the throttle valve and the sensor 144 which detects the rotational speed of gears, (i.e. the driving speed of the vehicle). The electromagnetic valve 108 will be de-energized and the electromagnetic valve 110 will be continuously energized. By this switching, the line pressure in the oil passage 280 will be supplied to the oil chamber 256 via the orifice and the oil passage 258, and also supplied to the oil chamber 276. The spool 240 will then move toward the right integrally with the spool 242 and stop as the spool 240 abuts against the stopper 246. The line pressure in the oil passage 172 will then be supplied to the oil passage 362 via the annular groove 264 to act on the pressure receiving surface 334 of the 1st-2nd speed shift valve 100 and on the pressure receiving surface 346 of the 4th speed clutch control valve 104. As a result, the spools 330 and 338 of respective valves 100 and 104 are caused to move toward the right end and the line pressure in the oil passage 238 will be supplied via the oil passage 364 to the oil chamber 366 of the kickdown brake 30 to move the rod 368 toward the left in resistance to the spring 370 to thereby engage the brake band (not shown) with the kickdown drum 52. On the other hand, the oil pressure in the oil passage 356 is discharged via the oil passage 226 and release the low reverse brake 32, and the 2nd speed is obtained.

At the time of the engagement of the kickdown brake 30, the duty ratio of the electromagnetic valve 106 is controlled by means f the electronic contr l device 112 by using the stroke detecting means 400, in such a

way that the oil pressure in the oil passage 238 (the il pressure to be supplied to the oil chamber 366 of th kickdown brake 30) is first maintained at the line pressure which is higher and reduced during gear shifting after a signal from the stroke detecting means 400 is obtained, in order to prevent shock.

Supplying high pressure oil until immediately before the engagement of the kickdown brake 30 reduces the time lag in engagement and thus shortens the time required for shifting without causing shock.

Gear shifting from 3rd to 4th speed also involves engagement of the kickdown brake 30. It is again possible to reduce the time lag and also the shock by supplying higher pressure oil until just before the brake 30 is engaged.

With regard to the stroke detecting means 400, there is no need to make any adjustment even if some parts such as the brake band are worn as long as the stroke detecting means 400 is constructed to always maintain the constant relation between the stroke L_1 and the gap L_2. It is to be noted, however, that other stroke detecting means such as a potentiometer can be used to directly detects the position of the rod 368.

As has been described in the foregoing by way of embodiments, the present invention is capable of reducing shock caused by gear shifting by controlling the oil pressure to be supplied to the frictional elements by an electronic control device. The construction is simple compared with other control devices of the oil pressure type.

Because the frictional elements, which are switched in accordance with the changes in the gear ratios, are actuated by a suitable overlap of torque capacities depending on the condition of the vehicle, shocks caused by such shifting can be alleviated. Further, since it enable to reduce the time which elapses from the start of the gear shifting to the beginning of the effective gear shifting, smooth and shockless shifting can be obtained without requiring a long overall time.

In each of the foregoing embodiments, the overlap regulating means was constituted by connecting the front clutch 24 and the oil chamber 382 to which the oil pressure is supplied when disengaging the kick-down brake 30, with the oil passage 376. But a hydraulic circuit without said oil passage 376 can be obtained by interposing a timing valve 31, as shown in the U.S. Pat. No. 3,832,915, in said hydraulic circuit and supply the oil pressure from the electric control device of the present invention to the port 63 of said timing valve 31, obtaining same effect as that of the present invention.

What we claim is:

1. A control device for an automatic transmission gear system having a change speed gear assembly with plural gear ratios between an input shaft and an output shaft, comprising:

first frictional element means actuated by fluid pressure to obtain one of said gear ratios for lower speed driving;

an actuating means for actuating said first frictional element means, having first fluid chamber means to which fluid pressure is supplied to activate said first frictional element means, and second fluid chamber means from which fluid proportional to fluid pressure supplied to said first fluid chamber is exhausted;

second frictional element means actuated by fluid pressure to obtain another one of said gear rati ns for high speed driving;

third fluid chamber means to which fluid pressure is supplied to activate said second frictional element means;

a fluid passage which connects said second fluid chamber means with said third fluid chamber means;

an exhaust passage for exhausting fluid from said fluid passage and connected to said fluid passage when shifting from higher to lower speed driving;

an orifice with an unchangeable opening provided in said exhaust passage for regulating amount of fluid to be exhausted;

a synchronization detection means for detecting whether said gear ratios have been obtained;

switching valves for switching fluid passages leading to said frictional element means at a time of shifting between higher and lower speed driving;

a fluid pressure regulating means, having a fluid pressure control solenoid valve, for controlling sum of transmittable torque of said first and second frictional element means by modulating the fluid pressure supplied to said first fluid chamber means during shifting from higher to lower speed driving;

an electronic control device for controlling said fluid pressure control solenoid valve to control said sum of transmittable torque in such a way that, said sum of transmittable torque is decreased when disengagement of the second frictional element means is detected by said synchronization detection means and said sum of transmittable torque is increased when engagement of the first frictional element means is detected by said synchronization detection means.

2. A control device for an automatic transmission gear system as claimed in claim 1, wherein said synchronization detection means has two sensor means for detecting rotational speed of two arbitrary rotational elements among plural rotational elements of said change speed gear assembly.

3. A control device for an automatic transmission gear system as claims in claim 2, wherein said change speed gear assembly is a ravigneaux-type planetary gear mechanism, and said two arbitrary rotational elements are a sun gear which becomes a reaction force element when said lower-speed gear ratio is obtained, and an annulus gear connected to the output shaft.

4. A control device for an automatic transmission gear system having a change speed gear assembly with plural gear ratios between an input shaft and an output shaft, comprising:

first frictional element means actuated by fluid pressure to obtain one of said gear ratios for lower speed driving;

an actuating means for actuating said first frictional element means, having a fluid pressure operated piston, a first fluid chamber on one side of said piston to which fluid pressure is supplied to activate said first frictional element means, and a second fluid chamber on another side of said piston from which fluid proportional to fluid pressure supplied to said first fluid chamber is exhausted;

second frictional element means actuated by fluid pressure to obtain an ther one of said gear ratios for higher speed driving;

a third fluid chamber to which fluid pressure is supplied to actuate said second frictional element means;

a fluid passage which connects said second fluid chamber with said third fluid chamber;

an exhaust passage for exhausting fluid from said fluid passage when shifting from higher to lower speed driving;

an orifice with an unchangeable opening provided in said exhaust passage for regulating amount of fluid to be exhausted;

a synchronization detection means for detecting whether said gear ratios have been obtained;

switching valves for switching fluid passages leading to said frictional element means at a time of shifting between higher and lower speed driving;

a fluid pressure regulating means, having a fluid pressure control solenoid valve for controlling sum of transmittable torque of said first and second frictional element means by modulating the fluid pressure supplied to said first fluid chamber during shifting from higher to lower speed driving;

a stroke detecting means for detecting a stroke position of said fluid pressure operated piston;

an electronic control device for controlling said fluid pressure control solenoid valv to control said sum f transmittable t rque in such a way that, said sum of transmittable torque is decreased by reducing said fluid pressure supplied to said first fluid chamber until said stroke detecting means detects that the piston is in the position just about to engage the first frictional element means during gear shifting from higher to lower speed driving, and said sum of transmittable torque is increased when engagement of the first frictional engagement element means is detected by said synchronization detection means.

5. A control device for an automatic transmission gear system as claimed in claim 4, wherein said change speed gear assembly is a ravigneaux-type planetary gear mechanism, and said synchronization detection means having means for detecting a rotational speed of a sun gear which becomes a reaction force element when said lower-speed gear ratio is obtained.

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